DEVELOPMENT OF A CONDENSING HEAT CELL BASED IN PLATE HEAT EXCHANGER TECHNOLOGY

Bosch Termoteknik Isitma ve Klima Sanayi Ticaret Anonim Sirketi

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À Helena e ao David

Resumo

O mercado das caldeiras de condensação a gás tem registado crescimentos significativos, em parte devido a legislações que favorecem a utilização de produtos mais eficientes.

Este crescimento introduziu uma grande competição entre os fabricantes, em especial ao nível da diferenciação dos seus produtos. Numa primeira fase a diferenciação incidiu na performance mas a tendência actual é baseada na redução do custo, mantendo ou melhorando a performance standard do mercado.

Esta erosão de preços promove a inovação e nesse sentido surge o conceito que origina a presente dissertação: desenvolver um permutador térmico de condensação baseado na tecnologia de permutadores de placas.

O presente trabalho tem como objectivo demonstrar a viabilidade deste conceito. Os objectivos foram definidos após uma análise cuidada do mercado e também do portfolio actual da empresa. O objectivo para o custo representa uma redução de 20% face ao produto mais competitivo do portfolio actual.

Utilizando ferramentas de simulação, foi efectuada uma primeira optimização do conceito, antes da sua validação experimental.

Os resultados experimentais confirmam as actividades de simulação e demontram que a performance térmica, emissões de gases poluentes e níveis de conforto estão em linha com os objectivos e os produtos de topo no mercado. A validação do conceito confirmou também que há necessidade de trabalho adicional ao nível da robustez do permutador térmico, dado que o conceito actual apresenta temperaturas dos vedantes críticos acima do aceitável, comprometendo a sua durabilidade.

Os resultados obtidos são, no entanto, promissores e já demonstram o potencial do conceito para industrialização.

Abstract

The gas condensing boilers market has registered a huge growth in the latest years, partly due to legislation that seaks to introduce more efficient products in the markets.

This growth introduced even higher competition between major suppliers, especially in features differentiation of their products. In the first phase, the emphasis was given to performance but the current trend is to offer the same or better performance at the lowest cost possible.

This price erosion promotes inovation. In that sense, it appears this concept behing the present dissertation: to develop a condensing heat cell based in the plate heat exchanger technology.

The present work intends to demonstrate the feasibility of this concept. The targets were defined after a carefull market and company current portfolio analysis. The target for cost reduction is 20% less than the most competitive product in current portfolio.

Using simulation tools, a first optimization for the concept was done, and lattely subjected to experimental validation.

The experimental results confirm the simulation activities and demonstrate that thermal performance, flue gas emissions and confort levels are in line with the objectives and top products in the market. The concept validation also shows that there is the necessity of extra work to improve the robustness of the heat cell. The current concept presents high temperatures in critical seals, that would lead to earlier failures.

The achieved results are, however, quite promising and demonstrate the high industrialization potential of the concept.

Acknowledgements

I have taken hard work in this novel and demanding project. However, it would not have been possible without the kind support and help of several individuals and organizations. I would like to extend my sincere thanks to all of them.

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CO_{DAF}	Carbon monoxide of the dry air free combustion products, (%)
$CO_{Measured}$	Measured concentration of carbon monoxide, (%)
CO_2	Carbon dioxide concentration, (%)
$CO_{2\,MAX}$	Maximum carbon dioxide concentration of the dry air free combustion products, (%)
$CO_{2\,Measured}$	Measured concentration of carbon dioxide, (%)
c_P	Specific heat capacity, (kJ/(kg.K))
d	Relative density of the test gas, (-)
D	Specific Rate, (l/min)
d_i	Water rate delivered, (l/min)
D_i	Domestic water rate corresponding to a mean temperature rise of 30 K that the boiler can supply in two successive delivery periods, (l/min)
D_C	Kitchen specific rate, (l/min)
d_r	Relative density of the reference gas, (-)
d_H	Hydraulic diameter, (mm)
f_i	Performance factor, (-)
F	Overall performance factor, (-)
H_2O	Water concentration, (%)
Hi	Net calorific value of dry reference gas at 15 ºC, 1013.25 mbar, (MJ/m^3)
I	Turbulence intensity, (%)
k	Thermal conductivity, (W/(m.K))
m	Corrected quantity of water, (kg)
$m_{i(10)}$	Quantity of water collected with a minimum temperature rise of 30 K, (l)
n	Number of tappings, (-)
N_2	Nitrogen concentration, (%)
$NO_{X\,DAF}$	Nitrogen oxides of the dry air free combustion products, (%)

$NO_{X\,Measured}$	Measured concentration of nitrogen oxides, (%)
O_2	Oxigen concentration, (%)
p_a	Ambient pressure at the time of the test, (mbar)
p_g	Gas pressure at the meter, (mbar)
P_{GAUGE}	Average pressure, (mbar)
Q_C	Corrected heat input (1013.25 mbar and 15 ºC, dry gas), (kW)
$Q_{Electric}$	Electrical consumption during the tapping efficiency test, (kWh)
Q_{gas}	Daily energy contributed calculated using H_i, (kWh)
Q_{H2O}	Useful energy recovered by the water, (kWh)
Q_{TOT}	Total delivered energy of used tapping cycle, (kWh)
T_1	Return temperature to the heat cell, (ºC)
T_2	Flow temperature from the heat cell, (ºC)
t_i	Tapping duration of the useful water, (min)
t_g	Temperature at the meter, (ºC)
$T_{GAS\,IN}$	Inlet temperature of hot flue gases in heat exchanger, (K)
t_m	Waiting time, (s)
$T_{Max\,Registered}$	Maximum registered temperature of heat exchanger or seal surfaces, (ºC)
T_{OUT}	Backflow temperature, (K)
t_s	Temperature stabilization time, (s)
$T_{WATER\,IN}$	Inlet water temperature in heat exchanger, (K)
V	Measured volumetric gas rate, (m^3/h)
V_g	Gas consumption, (m^3)
$V_{r(10)}$	Gas consumption measured and corrected to 15 ºC, 1013.25 mbar, (m^3)

List of Greek Symbols

ΔT	Mean temperature rise of the collected water, (K)
ΔT_1	Temperature variation according to water rate, (K)
ΔT_2	Temperature fluctuation at constant water rate, (K)
ΔT_3	Temperature fluctuation between successive deliveries, (K)
ΔT_i (t)	Instantaneous temperature rise during the tapping, (K)
μ	Dynamic viscosity, (m^2/s)
η_u	Useful efficiency, (%)
ρ	Density, (kg/m^3)

List of Acronyms

CFD	Computational fluid dynamics
CH	Central heating
CN	China
DHW	Domestic hot water
FEA	Finite elements analysis
DE	Germany
FR	France
HP	Heat pumps
LHV	Low heating value
NL	Netherlands
PHE	Plate heat exchanger
PT	Portugal
R&D	Research & Development
UK	United Kingdom

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1 General Introduction

Stringent customer and governmental demands are continuously shaping the water heating industry resulting in large product portfolio with new high efficiency boilers. The focus on the present study is given to gas condensing combination boilers, especially to their most important component: the heat exchanger.

The motivation behind this study is to provide a solution for the current market demands: high performance and comfort, while being environmentally friendly and especially being low cost.

Combining the current portfolio with a careful market analysis, that includes the technical and historical overview as well as major future trends, it will be possible to present very detailed targets to guide the project development. Outside the scope of the present work, but mandatory for a successful project, is the early recognition of risks and how to overcome them during the development phase.

A concept for condensing heat cell based on plate heat exchanger technology is firstly created and optimized using computational fluid dynamics. By doing so, the time required for such a development is greatly reduced. It also reduces the need to prototype innumerous samples that require special tools, which are time consuming and represent a very high development cost.

After this, an experimental validation is conducted with the most promising solutions derived from the simulation work.

2 Bosch

2.1 The BOSCH Group at a Glance

The Bosch Group is a leading global supplier of technology and services. In 2012, its roughly 306,000 associates generated sales of 52.5 billion (10^{12}) Euros. Since the beginning of 2013, its operations have been divided into four business sectors: Automotive Technology, Industrial Technology, Consumer Goods, and Energy and Building Technology [1].

	2009	2010	2011	2012	2013*
Sales revenue	38,174	47,259	51,494	52,464	46,400
Share outside Germany as percent	76	77	77	77	77
Number of associates (as of Jan. 1 of subsequent year)	270,687	283,507	302,519	305,877	281,000
located in Germany	111,710	113,557	118,776	119,232	107,000
located outside Germany	158,977	169,950	183,743	186,645	174,000
Capital expenditure	2,670	2,923	3,226	3,151	
Research and development cost [1]	3,348	3,073	4,190	4,787	> 4,500
Profit after tax	2,170	2,450	1,820	2,342	

Figure 1 - General Economic Data on Bosch Group Worldwide [2]

The Bosch Group comprises Robert Bosch GmbH and its approximately 360 subsidiaries and regional companies in some 50 countries. If its sales and service partners are included, then Bosch is represented in about 150 countries. This worldwide development, manufacturing, and sales network, is the foundation for further growth. Bosch spent some 4.8 billion (10^{12}) Euros for research and development in 2012, and applied for nearly 4,800 patents worldwide. The Bosch Group's products and services are designed to fascinate, and to improve the quality of life by providing solutions which are both innovative and beneficial. In this way, the company offers technology worldwide that is "Invented for life" [1].

The company was set up in Stuttgart in 1886 by Robert Bosch (1861–1942) as “Workshop for Precision Mechanics and Electrical Engineering.” The special ownership structure of Robert Bosch GmbH guarantees the entrepreneurial freedom of the Bosch Group, making it possible for the company to plan over the long term and to undertake significant upfront investments in the safeguarding of its future. Ninety-two percent of the share capital of Robert Bosch GmbH is held by Robert Bosch Stiftung GmbH, a charitable foundation. The majority of voting rights are held by Robert Bosch Industrietreuhand KG, an industrial trust. The entrepreneurial ownership functions are carried out by the trust. The remaining shares are held by the Bosch family and by Robert Bosch GmbH [1].

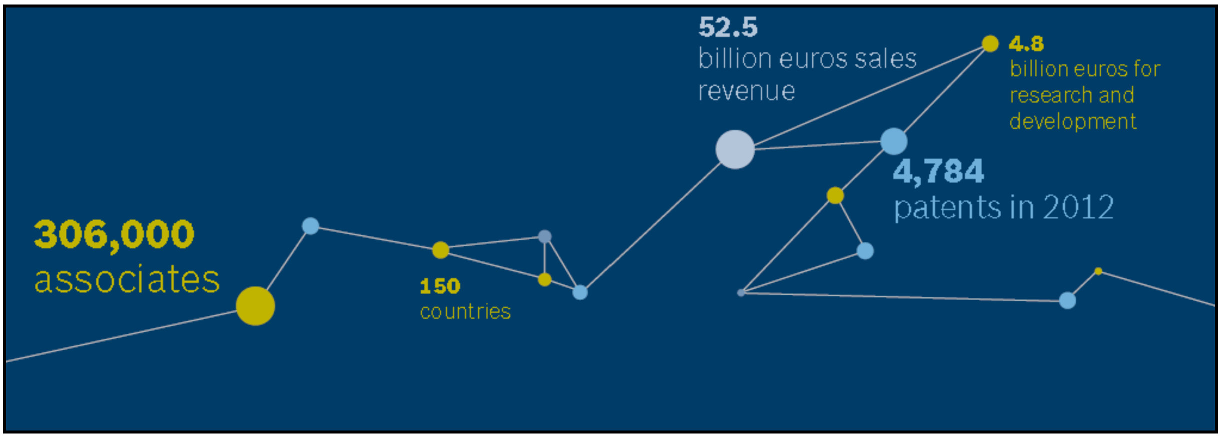

Figure 2 - General Information on Bosch Group Worldwide by end of 2012 [1]

Organization of the Bosch Group

Bosch Automotive Technology (UBK):

Automotive Technology is the largest Bosch business sector [3]. In 2012, it generated 59 percent of total sales. Its main business areas are: fuel-injection systems for internal combustion engines, peripheral devices for powertrain control, alternative drive concepts, active and passive vehicle safety systems, driver-assistance and other guidance functions, in car information and communication systems, and a range of after-sales, engineering-support, and service concepts for the automotive aftermarket.

GS - Gasoline Systems

DS - Diesel Systems

CC - Chassis Systems Control

ED - Electrical Drives

SG - Starter Motors and Generators

CM - Car Multimedia

AE - Automotive Electronics

AA - Automotive Aftermarket

ZF - Steering Systems (Affiliate)

Bosch Industrial Technology (UBI):

In 2012, Industrial Technology accounted for some 15 percent of total sales. Since 2013, Industrial Technology has comprised two divisions: Drive and Control Technology, a globally operating supplier for the mechanical engineering sector, with hydraulic components for mobile machinery, drive and control solutions for industrial applications, and components and systems for wind turbines. The second division, Packaging Technology, provides integrated packaging and process engineering solutions, above all for the pharmaceuticals, cosmetics, food, and confectionery industries.

DC - Drive and Control Technology

PA - Packaging Technology

Bosch Consumer Goods (UBG):

In 2012, the Consumer Goods and Building Technology business sector was responsible for nearly 26 percent of total sales. It was made up of the Power Tools, Thermotechnology, and Security Systems divisions, as well as the joint venture Bosch und Siemens Hausgeräte GmbH. Since the start of 2013, the Consumer Goods business sector has comprised the activities of Power Tools and the 50 percent share in the Bosch und Siemens Hausgeräte GmbH joint venture, which manufactures products for cooking, baking, cleaning clothes and dishes, cooling, and freezing.

PT - Power Tools

BSH - Household Appliances (www.bsh-group.com)

Bosch Energy and Building Technology (UBE):

Effective since 2013, Bosch has a new, fourth business sector: Energy and Building Technology. It includes the Thermotechnology division, a manufacturer of resource-conserving heating products and hot-water solutions, the Security Systems division, a provider of security technology as well as communication-center and other services for companies, and Solar Energy, a provider of photovoltaic solutions. As an energy services provider for commercial customers, the subsidiary Bosch Energy and Building Solutions GmbH develops integrated solutions that are eco-friendly, energy-efficient, and economical.

ST - Security Systems

SE - Solar Energy

TT - Thermotechnology

BEBS - Bosch Energy and Building Solutions GmbH

2.2 Bosch Thermotechnology

Bosch Thermotechnik GmbH stands for the Thermotechnology Division of the Bosch Group. With total sales of some three billion (10^{12}) Euros, it ranks among the world's leading suppliers of energy-efficient solutions for warm water and comfortable indoor climate.

As a systems supplier with a carefully coordinated product portfolio, it is in a position to meet the different requirements of the individual national markets as well as its customers' high demands.

The company maintains 20 plants in 10 countries, where it produces energy-efficient heating systems and hot water solutions for sale in some 50 countries worldwide. Its international strength derives from the presence in all major markets and the power of its brands. A great tradition and a high level of awareness - these are the characteristics of its brand strength [3].

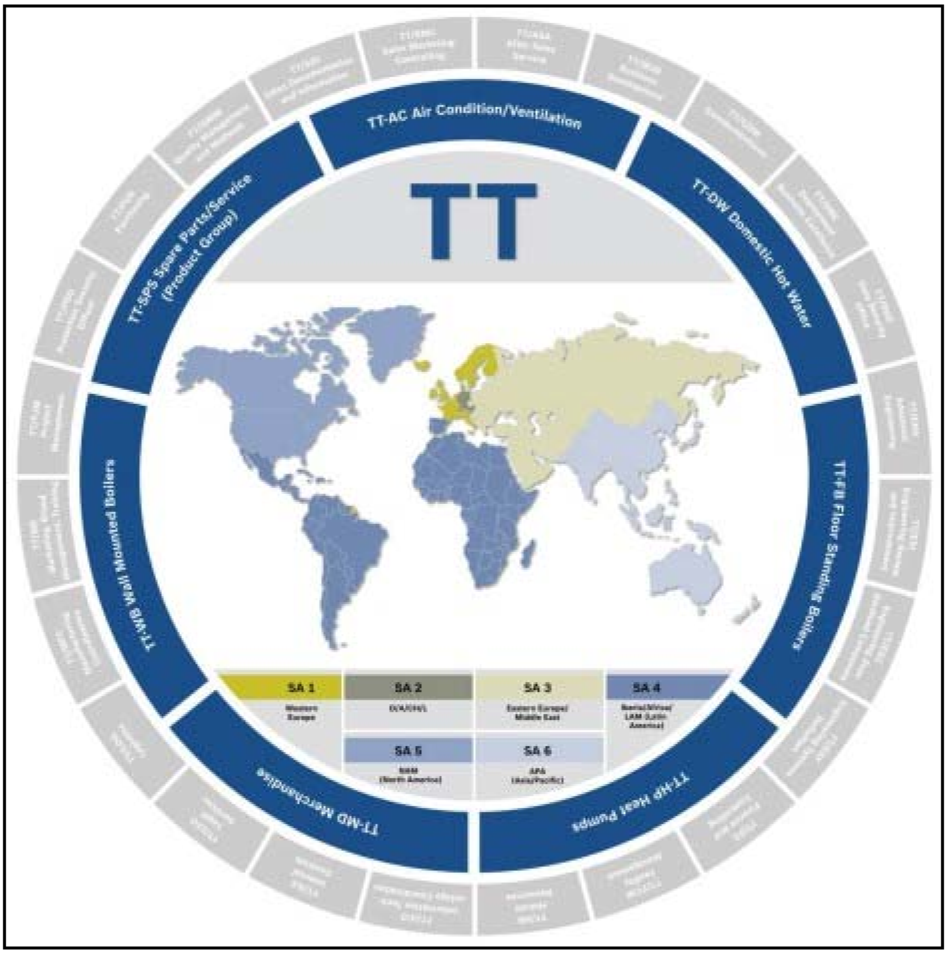

Figure 3 – Representation of Bosch TT Organization [3]

The TT-WB business unit covers conventional and condensing appliances with an output up to 100 kW. Also included are the compact heating units, controls and accessories for wall mounted heating appliances, as well the heat recovery ventilation.

New technologies as hybrids, which are wall mounted appliances combined with an electric driven heat pump, micro CHP appliances and gas driven heat pumps, are also part of the business unit.

Figure 4 – Technologies Developed and Commercialized by Bosch TT [4]

Products from the Wall Mounted Boilers business unit are manufactured at eight production sites around the world, employing a total of more than 3,000 employees [4]:

- Deventer (NL – Netherlands);
- Drancy (FR – France);
- Manisa (TR – Turkey);
- Saint-Thégonnec (FR);
- Wernau (DE – Germany);
- Worcester (UK – United Kingdom);
- Aveiro (PT – Portugal);
- Malu (CN – China).

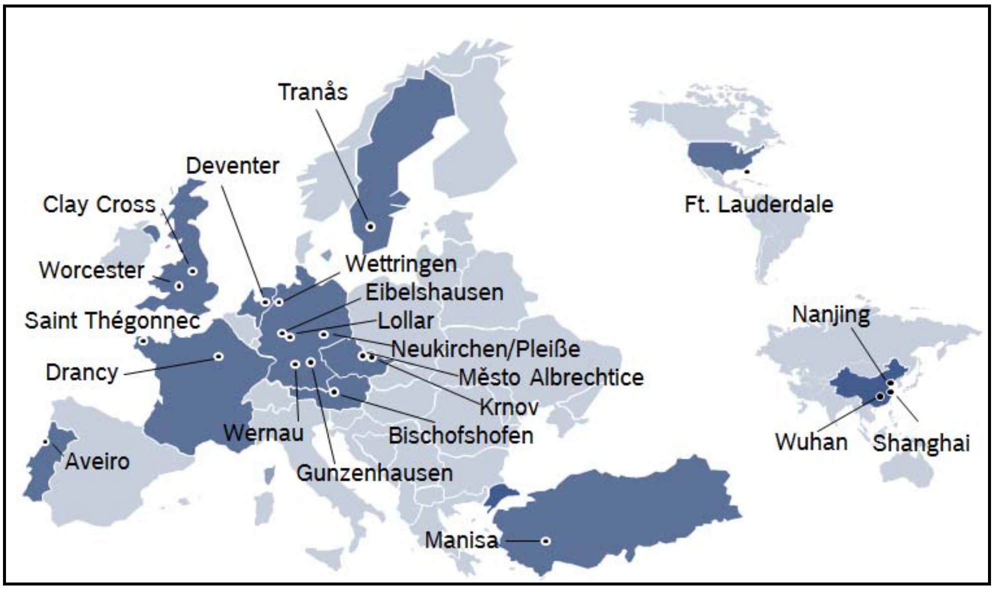

Figure 5 – Production locations of Bosch TT [4]

2.3 Bosch Turkey

Exporting to over 40 countries on 5 continents, carrying out 1.5% of Turkey's exports by itself with its 11,000 employees, Bosch Turkey recognizes sustainability as not only a management system, but also as a communication method that benefits to all parties in the markets which it operates. Bosch Turkey intention is to be a leader by means of the ethical way of doing business while considering its stakeholders' social, environmental, ethical and economic expectations. With its sustainability strategy, integrated with the sustainability strategy of global Bosch Group, Bosch Turkey develops policies and sets targets in line with the "Think globally, act locally" motto.

Bosch Turkey was founded in 1910, just 24 years after Bosch Worldwide. It has 4 locations representing 3 business units, and 3 official R&D centers supported by the Turkish Government. In 2013, its sales revenue reached 2.1 billion (10^{12}) Euros, representing 4.5% of Bosch Worldwide [3].

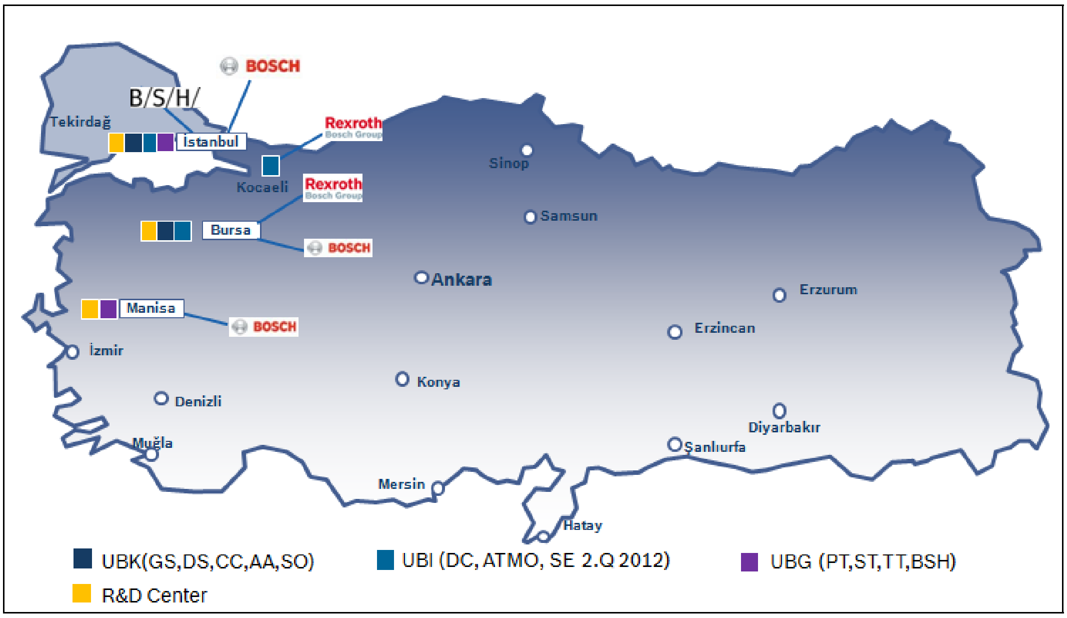

Figure 6 – Bosch locations in Turkey [4]

Bosch Turkey defines Sustainability as meeting the expectations of all stakeholders with an approach of right and ethical way of doing business, and by being one of the most respected brands of the world, acting responsibly along with integrating stakeholder expectations into the corporate priorities

Bosch Turkey, through product and service quality, innovation skills, power of creating employment, continuous improvement and efficiency experience, the range of environmentally friendly product and services, energy and waste management systems, strives to be above the global standards.

It also stands for being one of the leading companies of Turkey in the area of sustainable business practices. Bosch Turkey Sustainability Strategy, specified with a study during 2013, aims to present the strategic focus areas of Bosch Turkey and a transparent and clear map of its journey to all its key stakeholders regarding to sustainability as a dynamic and self-renewing company.

Bosch Turkey's Sustainability Strategy focuses on 4 key areas:

1. Environmental Impact;
2. Supply Chain;
3. Local Employment;
4. Stakeholder Engagement [6].

2.4 Bosch Thermotechnology Turkey

Bosch Thermotechnology Turkey which operates in heating, air-conditioning, and ventilation technologies, carries out distribution and marketing of heating, hot water, air-conditioning, and ventilation products of Bosch, Buderus, LG Air Conditioning Systems, and other leader brands in Turkey, Middle East, and Caucasus.

Bosch Thermotechnology Turkey respects nature and productive use of energy with the vision of creating values, sharing values, being the leader in the sector in both Turkish market and other operating countries. It adopts usage of durable, highly efficient, eco-friendly high-tech which consists of renewable energy and provides energy saving as the company mission.

The production site was awarded three years in a row (2011, 2012 and 2013) as "TT Best Plant". Its R&D department was also granted an important award in 2012 (delivered in beginning 2014) by being awarded with best Turkey R&D Department in its operation sector, which includes HVAC.

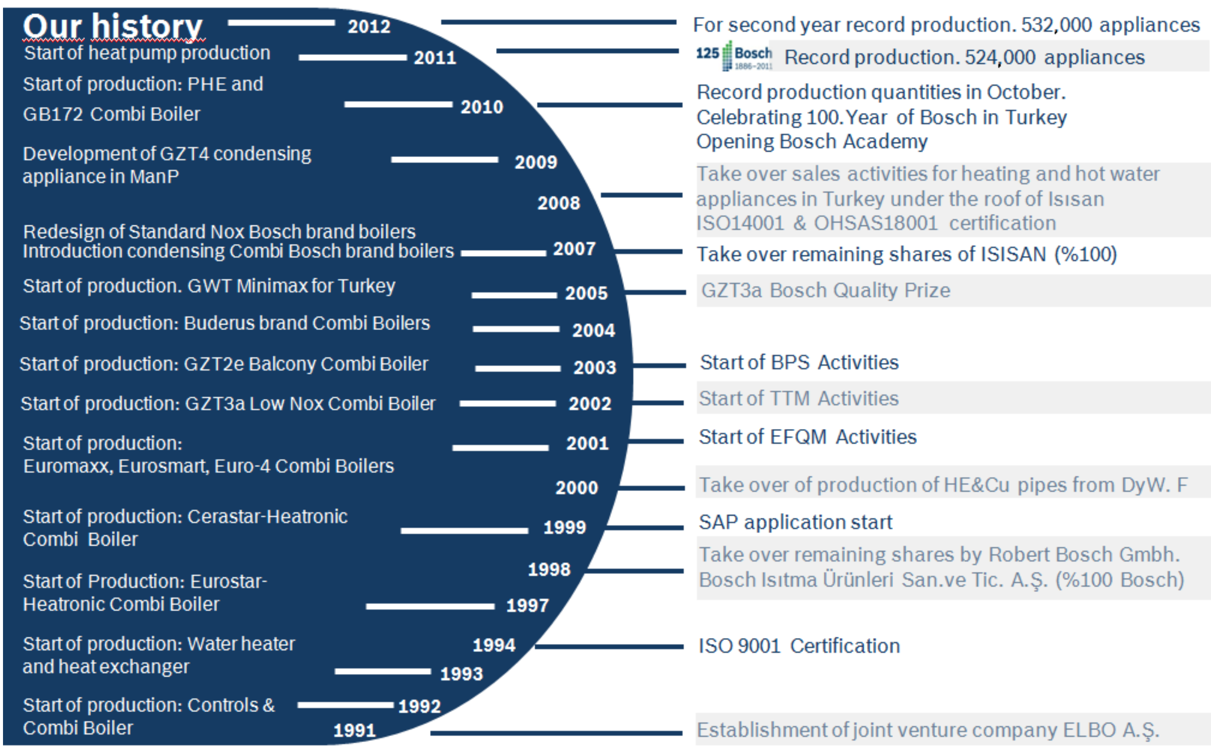

Figure 7 – Historic milestones of Bosch Thermotechnology in Turkey [4]

The range of products includes the following:

1. Heating systems; Boilers, wall-mounted condensing boilers, hermetical heater and other central heating systems;
2. Air-conditioning systems; Individual, commercial, and industrial air-conditioning systems;
3. Vapor systems;
4. Solar power and Heat Pump systems;
5. Heating equipment;
6. Ventilation systems.

Bosch Termoteknik Manisa plant is one of the most important heating product manufacturers in Turkey; which exports most of its production to Europe. Since 1992, Bosch Termotechnik is producing conventional (46%) and condensing (54%) combi boilers, gas-water heaters and components. Its production quantities are increasing year by year, having reached historical maximum of 600000 combi boilers in 2014 [4].

Manisa plant with its production, recruitment and exports as well as the contribution to the social and cultural development, adds value to Manisa and Turkish economy. Also, with its corporate structure, quality and service philosophy, designs technology for a better life, offering durable products. It has more than 750 associates, 70% of them directly connected to production facilities [4].

As one of the Turkish market leader in domestic heating and hot water systems, it is a name that stands for reliability, quality, efficiency and value for money. Although primarily destined for the European market these appliances are also sold in Turkish domestic market, where its market share is increasing year by year, especially with customized products for the region [4].

In Manisa plant lays an official Research and Development department. The department name stands for Thermo Technology-Wall Mounted Boilers/Engineering Platform and Application Development in Manisa. It is under the responsibility of TT-WB/NE, which is, within the business unit wall-mounted boilers, responsible for the development departments.

The development process is distributed over several sites, namely Deventer (NL), Drancy / St Thegonnec (FR), Manisa (TR), Wernau (DE), and Worcester (UK) [4].

TT-WB/NE-GC departments are responsible for the platform engineering of TT-WB sub-systems and components for wall-mounted condensing gas fired technology comprising of: Heat cells and their components including fans, gas valves, burners, and heat exchangers; hydraulics and its components including pumps, plate heat exchangers and three-way valves are also developed in these departments. They also develop activities in combustion management systems, which are modules that optimize combustion to meet both the highest efficiency and the lowest emissions. TT- WB/NE-GC is therefore also responsible for the application engineering in Manisa as well as the competence for non-condensing platform appliances [3].

In Manisa plant, are developed wall-mounted boilers, systems and components for 36 markets all over the world with more than 24 appliance families, about 800 different appliance types. It is specialized both in conventional and condensing appliance platform development, hydraulics and building expertise in Plate Heat Exchanger (PHE) component technology. Water-water PHEs developed in this R&D center is currently in use in whole TT-WB plants bringing significant cost saving to entire TT system [3].

As of 2012, the development of premixed fan, hydraulic units and its components including pump, diverter valve, flow sensor, pressure relief valve has become one of its core competencies in addition to already existing development responsibility for all conventional appliance components since 2003 [3].

The department is currently enhancing its competence by collaborating with universities and increasing its focus on FEA (Finite Element Analysis), CFD (Computational Fluid Dynamics) and plastic simulations. In accordance with the strategy of increasing competency, currently the department supports several associates towards their advanced engineering degrees.

In addition to its development responsibility within TT-WB, as of 2011 it is giving application engineering support also to HP (Heat Pumps) business unit with dedicated headcounts.

The R&D Centre keeps its competitiveness via governmental grants received since 2011. Till the end of the Q2 2013, a total of 4.6 million Euros was received by incentives from the Turkish government [4].

TT-WB/ENG-Man department is divided into following subgroups employing total of 47 associates:

-ENG1-Man: Development and laboratory testing for appliances and electronics;

-ENG2-Man: Development and simulation for components and sub-systems;

-ENG3-Man: Design engineering and hydraulics;

-ENG4-Man: Project management, variants and RPP;

-ENG5-Man: Heat pump engineering applications;

-ENG-Man1: Engineering change management, documentation, certification, audits and BES.

The current dissertation is related with the development of a condensing heat cell under the responsibility of ENG2-Man group.

This group has the vision of "Providing innovative, reliable and sustainable component solutions, suiting customer requirements, with continuous technical support through competent employees."

Its mission is in accordance with BOSCH TT strategies and focus on:

- Innovative, competitive and reliable component development and their integration into end products;
- Supporting all development departments based on lessons-learned and know-how gained from theoretical, experimental and numerical studies;
- Following quality figures of the components systematically and taking actions to increase the end product quality;
- Increase technical competence by investing in advanced education of employees and create funding through incentive programs.

3 Wall-Hung Condensing Gas Boilers

3.1 Technology Overview

A domestic wall-hung boiler is a device that combines the instantaneous water production with space heating. It is composed mainly of the following components:

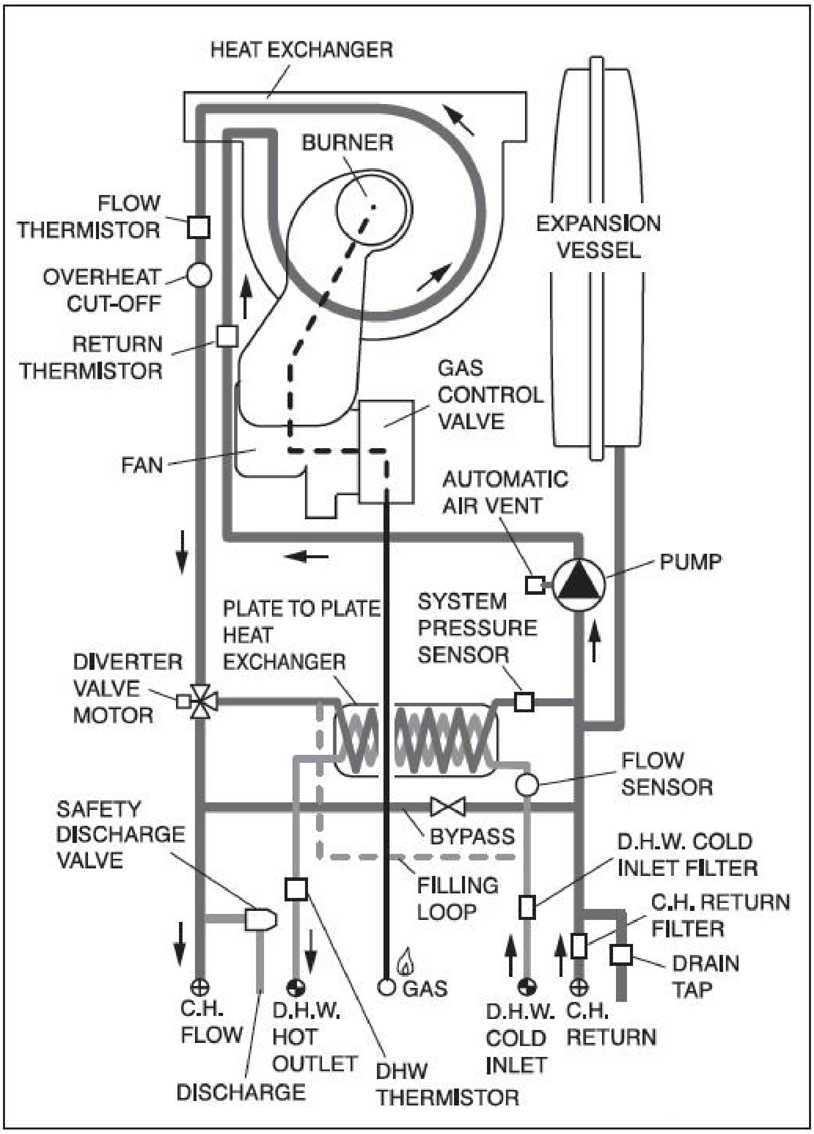

Figure 8 – Schematic layout of a combination boiler

Its operating modes are named DHW (Domestic Hot Water) and CH (Central Heating) and can be schematized as seen below:

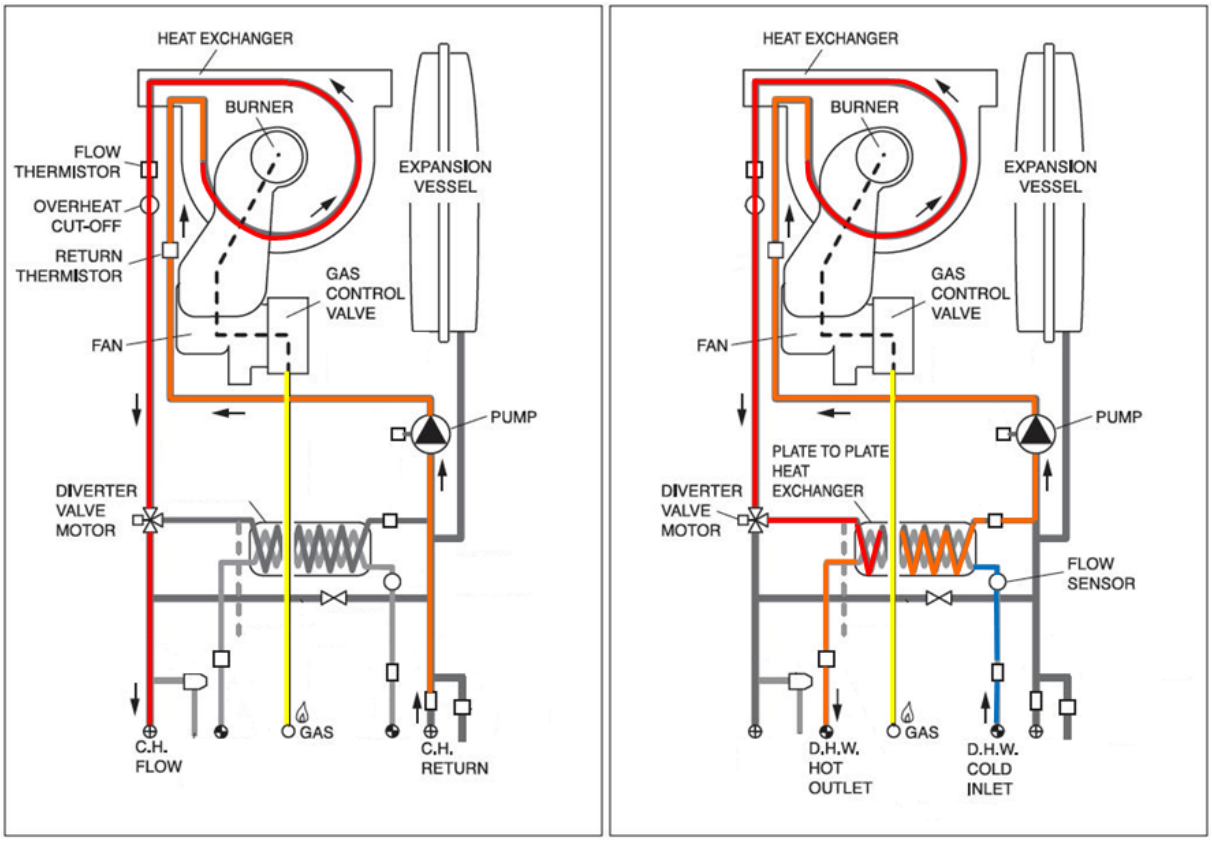

Figure 9 – Central Heating (CH) Mode (Left) and Domestic Hot Water (DHW) Mode (Right)

The domestic wall-hung boilers can be grouped into the three following main categories [5]:

Non-condensing boilers:

Typically, these boilers have atmospheric burners, copper heat exchangers and sheet metal chimneys (flues). The products of combustion (flue gases) are maintained at a sufficiently high temperature (resulting in low heat transfer efficiency) to allow them to exit the system using natural convection. If the flue gases do not contain enough heat to maintain proper stack action, the combustion products will spill back into the building. Forced draught boilers are also common in non-condensing boilers, especially to overcome the need of long flue pipes. In this case, they can also be room-sealed.

In addition, if the internal flue surface temperature is allowed to drop below the dew-point, moisture in the combustion products will condense on the internal walls of the heat exchanger and flues. The condensate is very acidic and will corrode the heat exchanger walls and damage metal and chimneys. By not capturing any latent heat from flue gases, non-condensing boilers operate at lower efficiency. Most boilers have a single combustion chamber enclosed by the waterways of the heat exchanger through which the hot gases can pass. These gases are eventually expelled through the flue, located at the top of the boiler, at a temperature of around 180°C. Therefore, its efficiency is usually limited to the range 84% - 87% (based on the LHV – Lower Heating Value – of the fuel). However, due to their relatively low cost of fabrication, they have still a strong presence in some markets.

Semi-Condensing Boilers:

Typically, near-condensing boilers use forced-draft solutions instead of atmospheric draft to pull gases through the heat exchanger. These boilers are usually equipped with an extra stainless steel or other corrosion-resistant material heat exchanger, since they are designed to tolerate the transient presence of condensate in the boiler, as during start-up. Because they have relatively high efficiency and relatively low flue gas temperatures, they require a flue construction that accommodates condensation downstream of the boiler. Otherwise, expensive and potentially dangerous stack damage may occur.

Condensing Boilers:

Typically, condensing boilers run at positive pressure with precisely-controlled premixed gas/air mixtures. All heat exchanger and flues surfaces are of corrosion-resistant material such as aluminum or high stainless steel grades due to the presence of corrosive gases and condensate (by design) in all heat transfer pathways.

Condensing boilers operate at higher efficiency by capturing some of the latent heat and virtually all of the sensible heat of combustion. In addition these boilers operate at high efficiency even at part-load conditions, when return water temperatures from space heating equipment are usually low. The efficiency range is between 98% in maximum under non condensing conditions, and 107.5% at minimum with low flow and return temperatures (values based on the fuel LHV).

3.2 Market Historical Overview and Future Trends

The total world domestic boiler market was estimated at US$ 10.9 billion and 10.46 million units in 2005 and is growing at a moderate rate over the last years [7]. The UK was and still is the biggest market in both value and volume terms.

Growth rates vary significantly between countries, with smaller markets generally offering higher growth potential. These are found in North Africa, the Middle East and Central Asia, with double-digit growth in Kazakhstan, Tunisia, Morocco, Jordan and Turkey up to 2010. China with its booming economy is set to grow at a rate of almost 7% a year. Mature markets such as the UK and Germany are expected to decrease very slightly over the next few years.

The relation of the individual market segments has been changing over the last years. Wall hung non-condensing boilers, which were representing about 50% of the total market in volume, are showing heavy decline in the latest years. Condensing boilers on the other hand, are growing rapidly. In some countries like UK, it is no longer possible to install conventional boilers.

The wall hung condensing boiler market is highly concentrated. The UK represents close to 50 percent of the total market, followed by the Netherlands and Germany which account for 16 and 15 percent respectively. Thus, around 80% of the total condensing market is accounted for by just three countries. The next biggest markets are Japan, South Korea and Italy.

The domestic boiler market is highly regionalized with the major players generally only strong in their home continent. China is an exception because leading European brands such as Bosch, Buderus and Viessmann hold important market shares there. Wall hung units are dominated by some nine major companies, although there are many more big brands used by these companies in individual countries.

The Asia Pacific region has many huge domestic boiler manufacturers, who have a significant share of the world market but are only strong in their home country. The exceptions are Rinnai and Noritz (Japan and China) and Navien and Kyungdong (Korea and China). North American boiler manufacturers are also confined to their own region, finding it difficult for existing products to conform to European regulations.

Below it is presented an individual summary of country market features:

China: The market is still dominated by larger commercial/industrial boilers including those for district heating. The market is forecast to experience strong annual growth of around 7% in volume.

France: France is the 4th largest market in the world by volume and 5th largest by value. The balance of power between oil and gas is largely determined by the differential in fuel prices which is a political decision in France.

Germany: The third world's largest market by value, it is the world number one for oil/pressure jet burning boilers and world number four for floor standing gas atmospheric boilers. German boiler manufacturers make boilers to a much higher specification than required by exports to other European countries, particularly for wall hung units.

Japan: This market is very difficult to understand for outsiders and stands at around 400,000 units. Boilers are regarded as part of the immense and highly sophisticated gas instantaneous market and are taking market share. The word 'boiler' confusingly applies to both markets.

Most boilers are the combination type and are used to supply under floor heating (and are known as 'floor heating boilers') and also provide hot water to sinks and to the unique Japanese 'bath circulator'.

The products are of very high quality with many unique features and importer technology will probably need to upgrade significantly in order to compete with local manufacturers. Included in the figures above are the rapidly growing 'hot water room heaters' or a 'split' boiler system. This is a kerosene boiler sold with a matched fan convector, not dissimilar in appearance and arguably akin to an air conditioning split system. Indeed, "multisplits" are possible with multiple fan convectors. Fujitsu developed the concept and it is the market leader.

Netherlands: The country has a very large market given its population size. The Netherlands is the second largest condensing wall hung boiler market worldwide. The Dutch condensing boiler market by far exceeds the non-condensing boiler market. The market is almost entirely based on gas burning appliances, with hardly any oil burning appliances found.

South Korea: The world's second largest market in volume and fourth largest in value. As well as the huge wall hung market, Korea is also easily the largest market for oil/pressure jet boilers, and totally dominates the world electric boiler market. Domestic boilers are almost entirely used with under floor heating (known as 'ondol' or the bottom piping market), used in 95% of homes. Korea is the world's number one market for under floor heating by a huge margin. Boilers are not surprisingly also known as 'ondol' units and are almost entirely the combination type.

United Kingdom: Wall hung condensing boilers have experienced a dramatic increase in 2005 when they more than doubled in units sold. This expansion came at the expense of equivalent non-condensing boilers. The reason for this shift was that building regulations have been introduced in England and Wales requiring the installation of condensing boilers in all new and replacement situations where a gas-fired boiler is installed. There is still a very strong installed base in the UK of houses with separate cold water tanks in the loft (attic) which supply both boilers and water heaters.

USA: The USA is dominated by ducted warm air furnaces with 3-4 million units typically sold per year, with most domestic boilers confined to the north east of the country.

Despite the above trend, the USA is the world's second largest market for floor standing gas atmospheric and for oil/gas pressure jet boilers.

Although in most of the countries there is a strong movement towards higher efficiency products, with combination condensing wall-hung boilers gaining more and more market share, there is strong price erosion: customers are demanding more features than ever at lower prices.

The trends are nowadays very clearly identifiable:

- System integration with renewable sources and heat pumps;
- Micro CHP, even though for the time being it is only a niche market;
- High end “in-boiler controls” with use of touch screens and enhanced by mobile control;
- High efficiency domestic water, especially in UK, Netherlands and Germany;
- Wide modulation ranges, starting in 1:8, nowadays the standard request, with some countries like Italy pushing very firmly to 1:10 and higher. In this aspect, the US is an exception for a long time, where a wide modulation range of 1:20 is standard for some years in the water heaters segment.

With the exception of system integration, micro CHP and high end controllers, well received in high price and quality markets like Germany, the others requirements are becoming standard, i.e. they must be inserted in the baseline of any supplier portfolio.

To achieve this, further cost improvements have to be done by suppliers. One of the biggest weights in a boiler price is the so called heat cell. This includes all the gas, air, gas/air mixture and flue paths. In other words, the components included are gas valve, premix fan and gas/air mixer, burner and burner hood, main heat exchanger, sump and flue pipe.

To reduce the weight of these components, strong efforts have to be done in gas/air mixing, but especially in the main heat exchanger. This is the target of the project described in the present dissertation.

4 Project Description

4.1 From the Idea to the Concept

The target of present project is to deliver a new heat cell concept that is cost driven. The target is to reduce material cost by at least 20% when comparing a 36 kW solution to products in current portfolio. The heat cell definition includes the following components:

1. Air Suction Pipe;
2. Gas Outlet Pipe;
3. Gas / Air Mixer;
4. Gas Valve;
5. Premix Fan;
6. Outlet Premix Fan Seal;
7. Burner Hood;
8. Burner Hood Seal;
9. Burner;
10. Burner Seal;
11. Heat Exchanger;
12. Igniter;
13. Igniter Seal;
14. Sump;
15. Sump Seal;
16. Flue Pipe Seal;
17. Flue Pipe

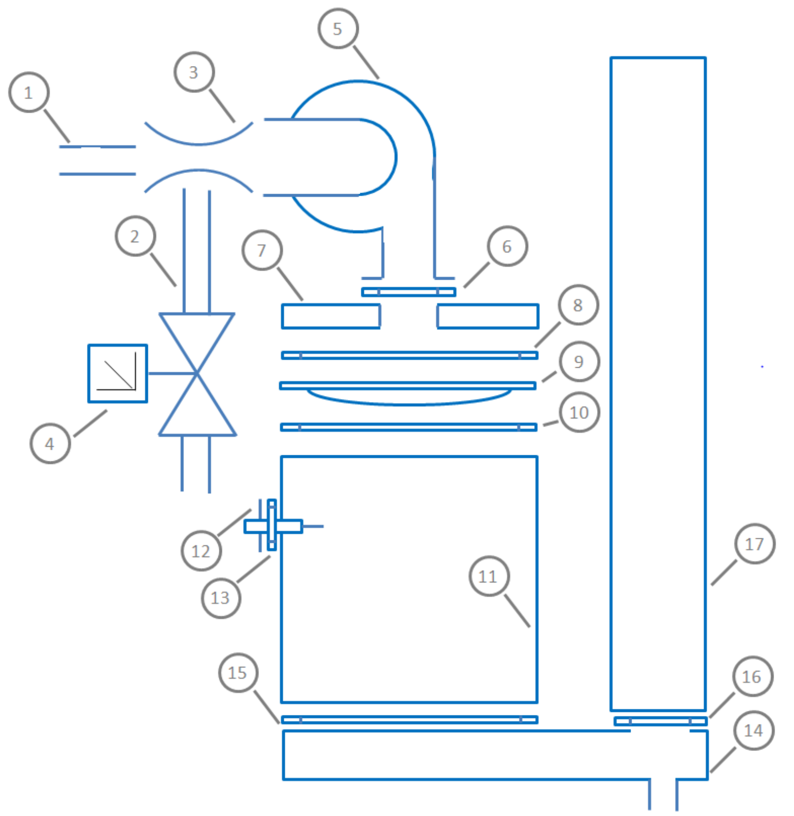

Figure 10– Schematic of components included in a Condensing Heat Cell

Additional requirements for the application are listed below:

- High power density (to be the best in class);
- Low weight and low water content to enable high response time and high DHW (Domestic Hot Water) efficiency;
- Scalable from 15 to 50 kW without designing a new heat exchanger;
- 36 kW heat cell with expansion vessel should fit the smallest 36 kW appliance casing in current portfolio (580×350×280 mm – appliance without expansion vessel);
- Standard mechanical modulation range of 1:8 easily upgraded to 1:10 (with electronic modulation).

The design elements above and their functions are presented in the system structure below:

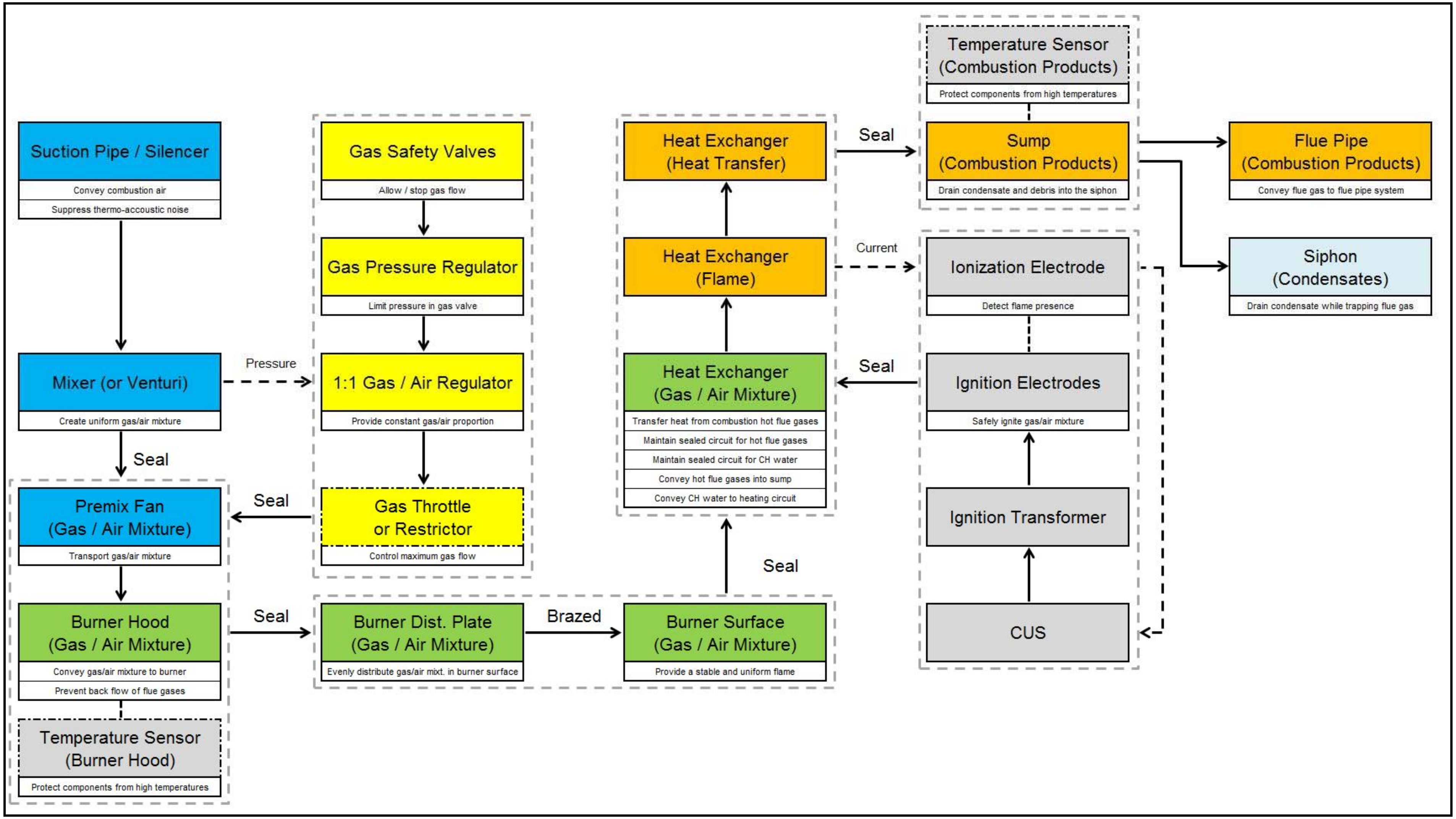

Figure 11 – System structure with Design Elements and Functions

The concept must be modular to allow further economical gains. This means that it must be possible to change heat exchanger dimensions when appliance power is changing. This is a key factor, as existing ranges of condensing boilers share a unique heat exchanger independent of power, i.e. it is dimensioned for the maximum power.

To achieve these requirements, it is proposed to proceed with a design based on plate heat exchanger technology – cross flow with un-mixed flows.

A plate heat exchanger (PHE) is a type of heat exchanger that uses metal plates to transfer heat between two fluids. This has a major advantage over a conventional heat exchanger in that the fluids are exposed to a much larger surface area because the fluids spread out over the plates. Each plate is made by stamping or embossing a corrugated (or wavy) surface pattern on sheet metal. This facilitates the transfer of heat, and greatly increases the speed of the temperature change. Over 60 different patterns have been developed worldwide but the typical plate geometries (corrugated patterns) [8] are shown in figure below:

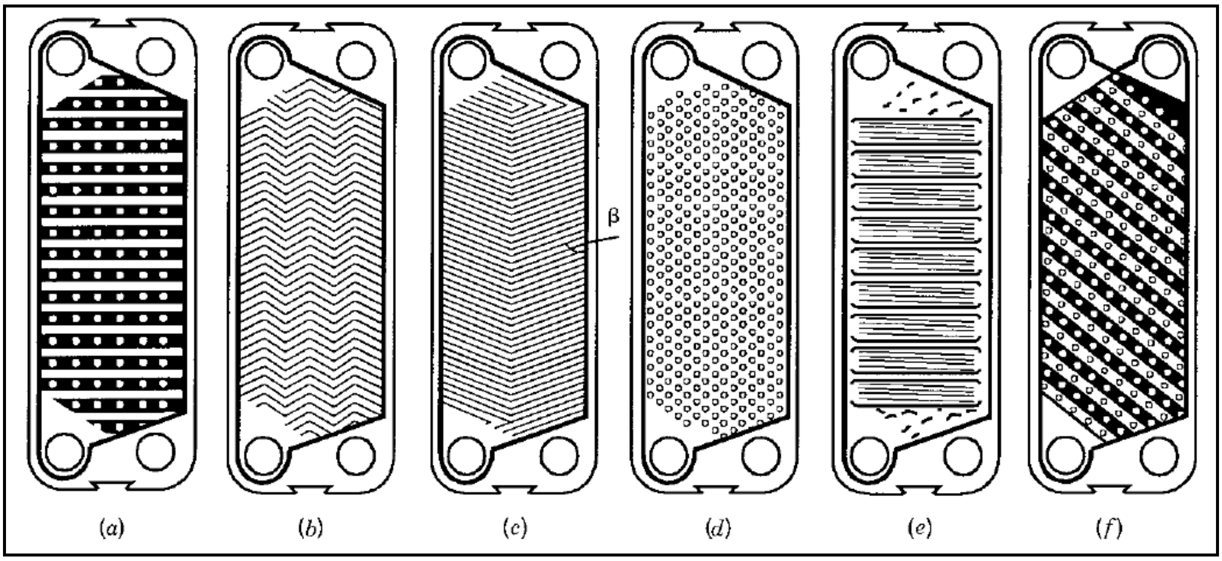

Figure 12 – Typical plate geometries used in PHE: (a) washboard; (b) zigzag; (c) chevron or herringbone; (d) protrusions and depressions; (e) washboard with secondary corrugations; (f) oblique washboard

The corrugations on successive plates contact or cross each other to provide mechanical support to the plate pack through a large number of contact points. The resulting flow passages are narrow, highly interrupted, and tortuous, and enhance the heat transfer rate and decrease fouling resistance by increasing the shear stress, producing secondary flow, and increasing the level of turbulence. The corrugations also improve the rigidity of the plates and form the desired plate spacing [8].

The heat transfer surface area can readily be changed or rearranged for a different task or for anticipated changing loads, through the flexibility of plate size, corrugation patterns, and pass arrangements. High shear rates and shear stresses, secondary flow, high turbulence, and mixing due to plate corrugation patterns reduce fouling to about 10 to 25% of that of a shell-and-tube exchanger, and enhance heat transfer. To avoid clogging, the largest suspended particle should be at most one-third the size of the average channel gap [8].

Additionally, the surface area required for a plate exchanger is one-half to one-third that of a shell-and tube exchanger for a given heat duty, thus reducing the cost, overall volume, and space requirement for the exchanger. Also, the gross weight of a plate exchanger is about one sixth that of an equivalent shell-and-tube exchanger

Considering the application under high temperature flue gases and acidic media condensates, the PHE has to avoid any kind of joints and therefore has to be brazed. A vacuum brazed plate heat exchanger is the choice for high-temperature and high-pressure duties, and it does not have gaskets, tightening bolts, frame, or carrying and guide bars. It consists simply of stainless steel plates, all generally copper brazed. Nickel brazing or protective coatings are solutions for corrosive environments. Since this exchanger cannot be opened or cleaned, its applications are limited to negligible fouling cases and low degree of corrosion.

Taking into consideration the requirements and the above mentioned intrinsic properties of a plate heat exchanger, it becomes evident that this technology is suitable to meet the demands for the application.

Based on this, the first concept was created: a cross flow plate heat exchanger (AISI 306L stainless steel plates brazed with copper foil) inside an aluminum casing with water cooling (see Picture 13 below for reference).

The central heating water enters in the aluminum casing, which is connected to the plate heat exchanger using a standard hydraulic connection (not visible on Picture 13 below).

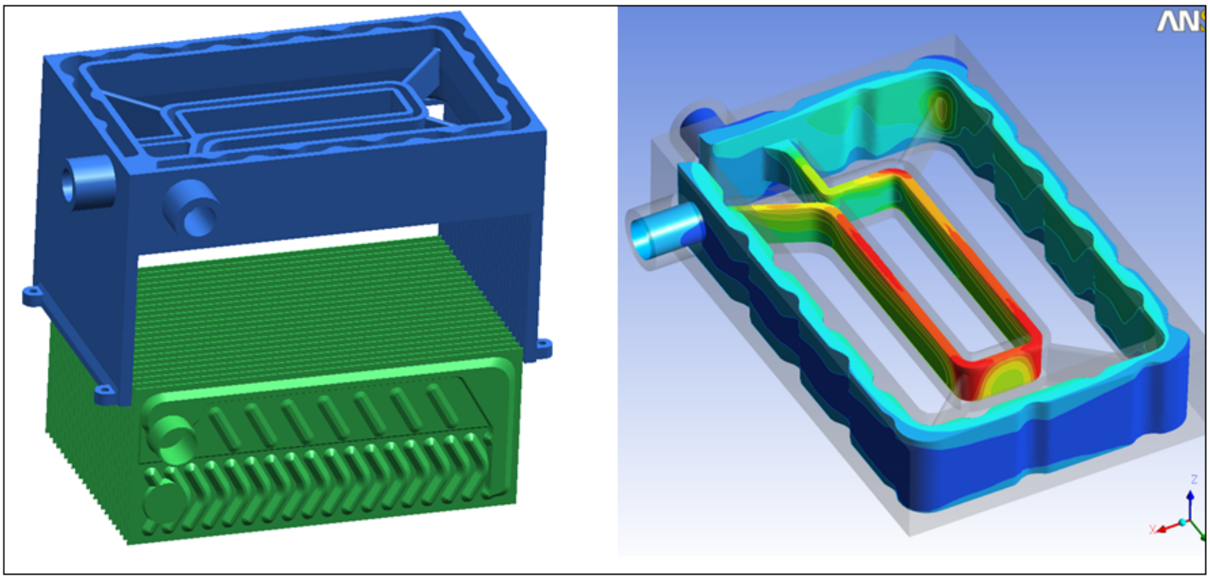

Figure 13 – First heat exchanger concept design

The above concept presents several disadvantages, which prevent it to be further analyzed:

- It is not completely modular. Even though the heat exchanger can easily be scalable by adding or removing plates, different appliance powers require different aluminum housings, therefore new tools and investment;
- The need of having a water cooled channel in the aluminum housing, creates higher pressure drop on the water side;
- The above mentioned water channels require seals with high temperature resistance, increasing cost and creating additional risks during lifetime;
- The design does not prevent flow from passing between the heat exchanger and the side walls, therefore resulting in efficiency loss.

To prevent the above difficulties, the following new concept was created. This design creates a completely closed combustion chamber, without additional seals, using the plate design to achieve it. The water modules of the heat exchanger are arranged in parallel flow. The flue gas and water flows are perpendicular to each other.

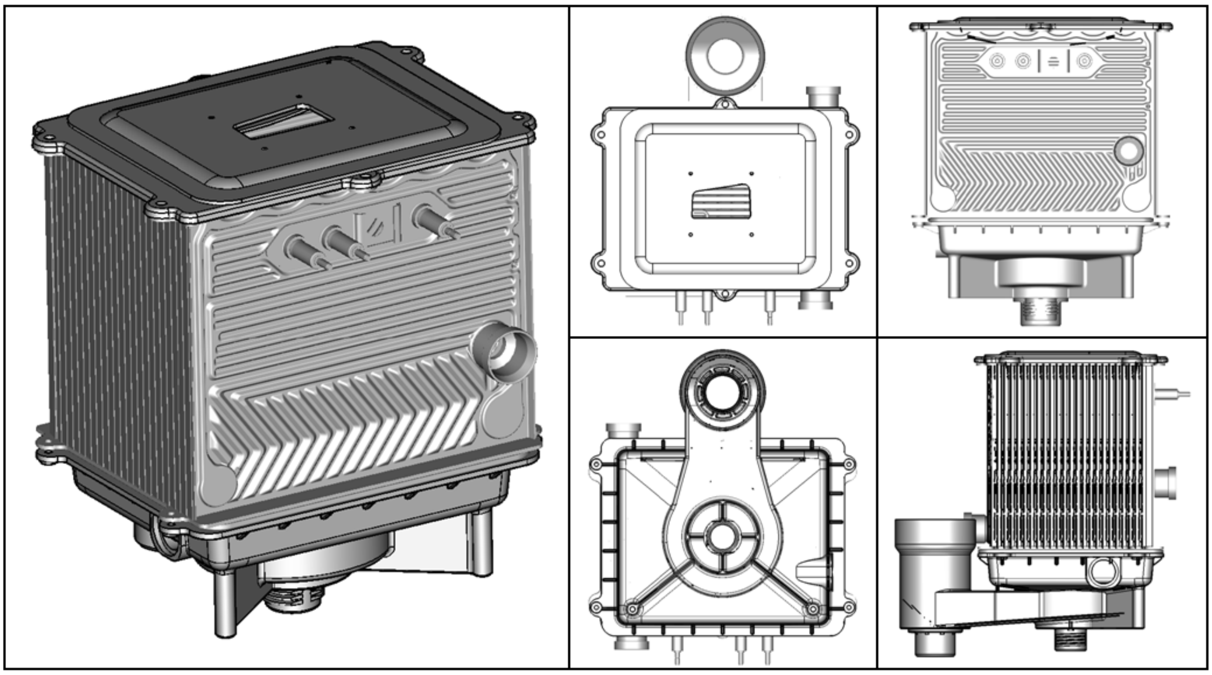

Figure 14 – Second heat exchanger concept design – external views

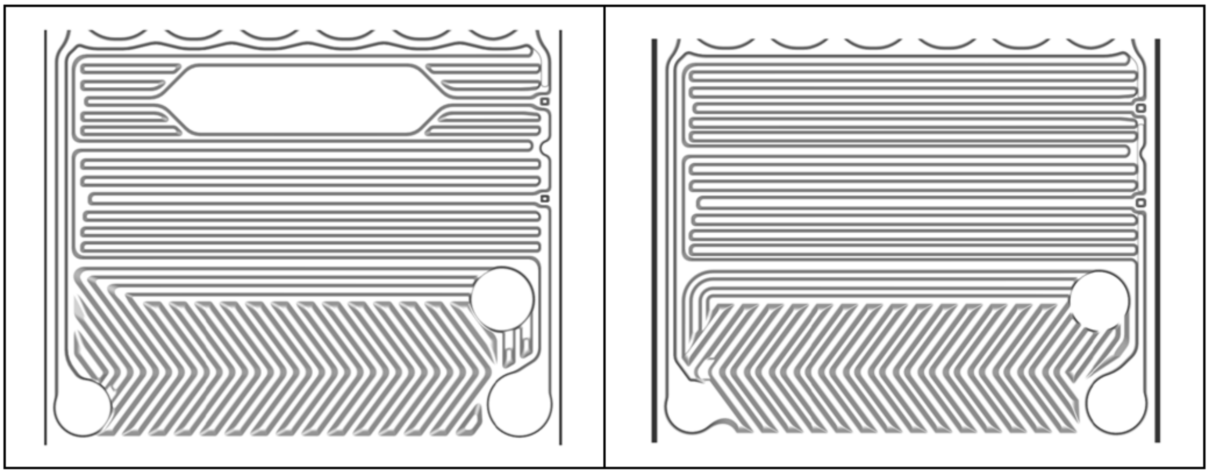

Figure 15 – Second heat exchanger concept design – front (left) and back plate (right) design

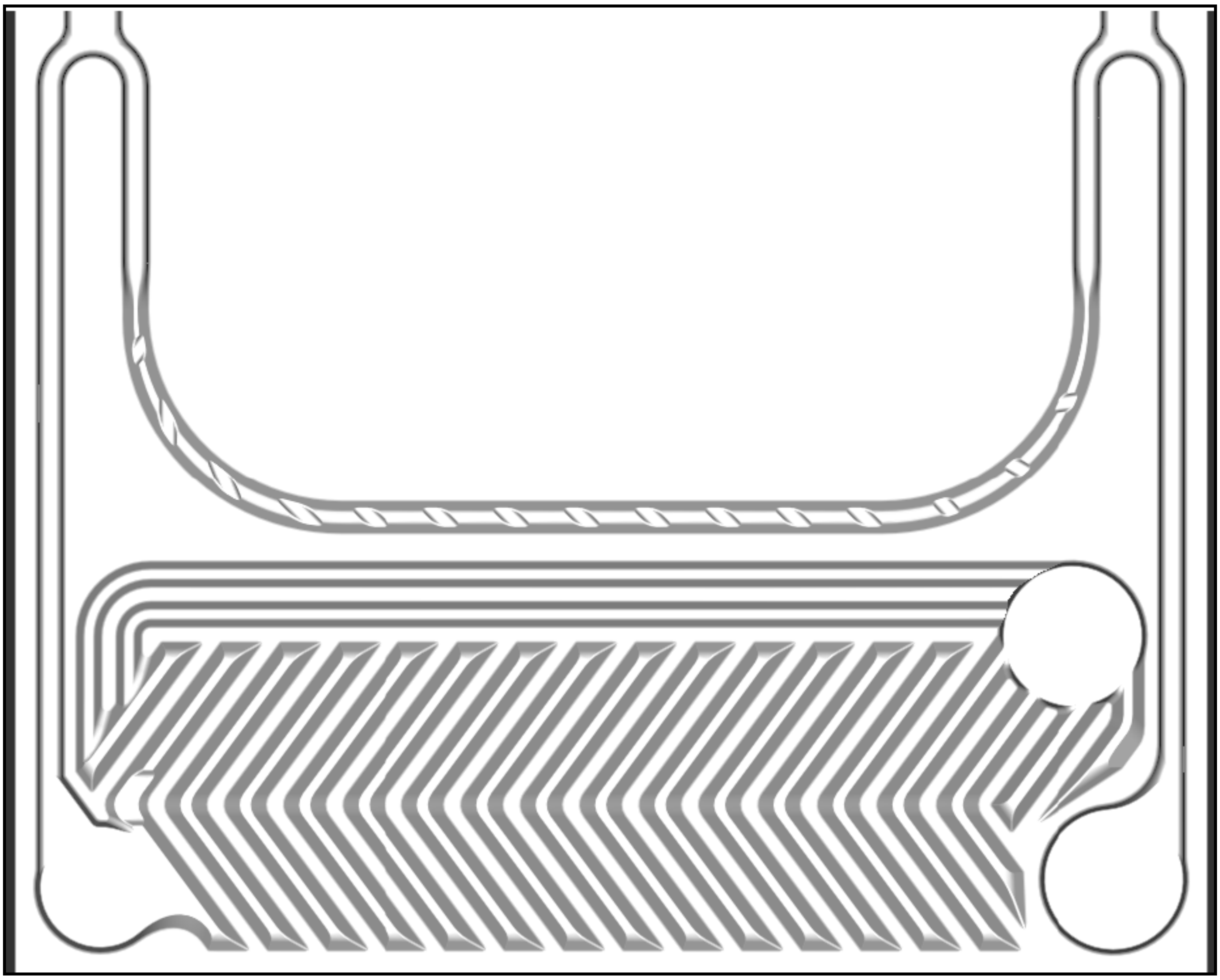

Figure 16 – Second heat exchanger concept design – middle plates design

As a summary, the System Elements and System Diagram are presented:

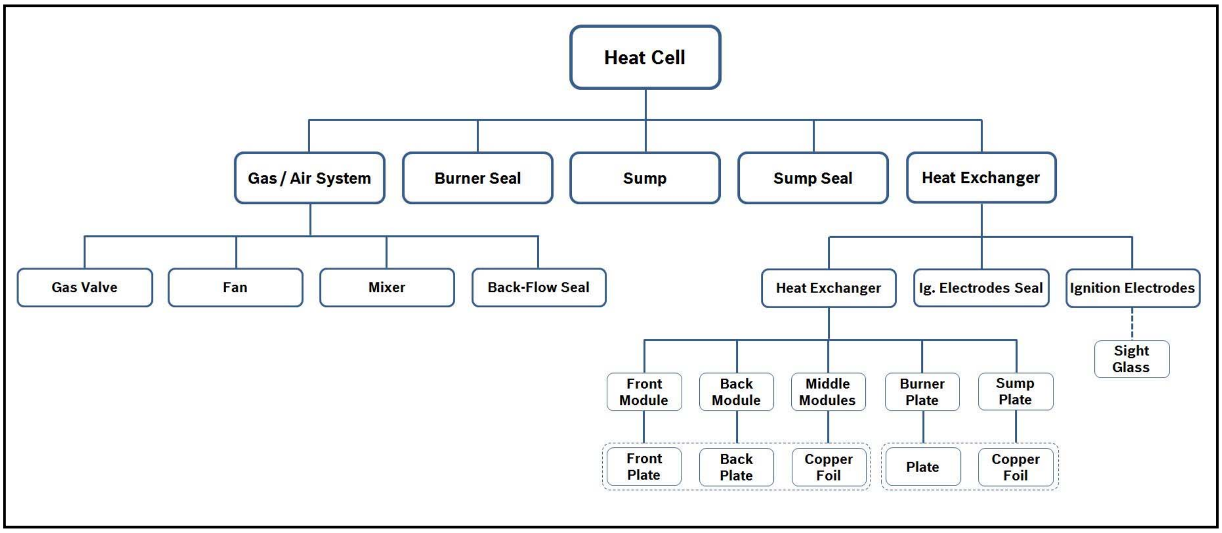

Figure 17 – Second heat exchanger concept design – system elements

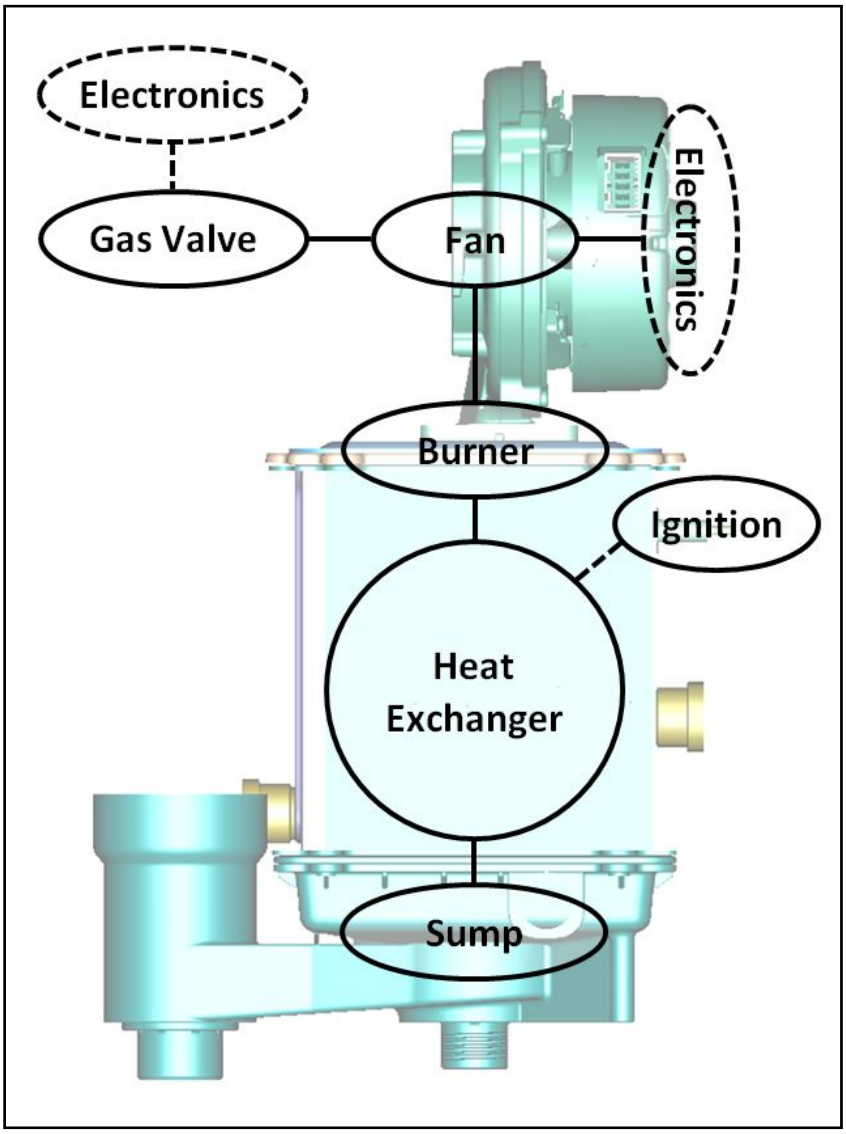

Figure 18 – Second heat exchanger concept design – system diagram

4.2 Main Technical Targets

The main technical targets for the heat cell can be summarized in the table below:

BOSCH							Requirement list
Requirements					**Own specification**		
Requirement group	Number	Requirement	Value range	Unit	KPI	Specification	Remark
Basic Function	1.1	DHW Maximum Heat Input	36.0	kW		36	
	1.2	CH Heat Input	-	kW		24.6	
	1.3	Minimum Heat Input	3.6 to 7.2	kW		7.2	
	1.4	Modulation Range	1:5 to 1:10	-		1:5	Higher modulation range possible with CMS ready
	1.5	Max. CH Flow Temperature	82	°C		82	
Installation	2.1	Flue Length Horizontal 60/100	6	m		6	
	2.2	Flue Length Horizontal 80/125	15	m		15	
	2.3	Maximum condensate rate	-	l/h		2.0	
	2.4	pH of condensates	-	-		≥ 5.5	
Construction	3.1	PMAT Reduction	≥ 20%	€	X	≥ 20%	
	3.2	Pressure Drop CH water @ (1500 l/h)	Best in class	mbar		≤ 120	
	3.3	Pressure Drop Flue Gas Side	-	Pa		-	min. 36kW (Hi) CO_2 = 9.1 ± 0.2% with current TT Fan
	3.4	Noise output level (Max. central heating)	-	dB(A)		< 45	
	3.5	Appliance Size (h×w×d))	580×350×280	mm		580×350×280	Same size as LCN but including expansion vessel
	3.6	Space Envelope Heat Cell with Fan and Sump (h×w×d)	-	mm		440×235×275	
	3.7	Space Envelope Heat Exchanger with Connections (h×w×d)	-	mm		170×215×195	
	3.8	Volume Heat Exchanger with Connections	Best in class	dm^3	X	7.0	
	3.9	Specific Power Density	Best in class	kW/dm^3		5.2	
	3.10	Heat Cell Weight (without fan)	Best in class	kg	X	≤ 7	
	3.11	Primary Water Content	Best in class	l		≤ 1	
Efficiency	4.1	Tapping Efficiency according EN13203-2 (cycle nr. 2)	> 72.5	%		> 72.5	
	4.2	Sedbuk 2005	A	-		A	Applicable to UK only
			≥ 90	%		≥ 90	
	4.3	Sedbuk 2009	≥ 90	%		≥ 90	

	4.4	Gaskeur HR (High Efficiency Heating)	-	%		≥ 107	Applicable to NL only
		Gaskeur HRww (High Efficiency Warm Water)	-	-		Yes	
		Gaskeur CW (Comfort Warm Water)	-	5		5	
		Gaskeur SV (NO_X Emissions)	-	< 160		< 160	
		Gaskeur SV (CO Emissions)	-	< 40		< 40	
		Gaskeur NZ (Solar Compatible)	-	-		-	
	4.5	DHW Max Rate Heat Output	-	kW		35.3	
	4.6	CH Max Rated Ouput 8060	≥ 24	kW		24	
	4.7	CH Max Rated Ouput 5030	-	kW		25.35	
	4.8	CH Max Rated Ouput 4030	-	kW		25.60	
	4.9	Flue Gas Temperature 8060 Max.	-	ºC		< 80	
	4.10	Flue Gas Temperature 4030 Min.	-	ºC		< 35	
	4.11	Efficiency at CH Max (8060 condition)	≥ 97.5	%		≥ 97.5	
	4.12	Efficiency at CH Max (5030 condition)	≥ 103	%		≥ 103	
	4.13	Efficiency at CH Max (4030 condition)	≥ 104	%		≥ 104	
	4.14	Efficiency at CH Min (5030 condition)	≥ 107.5	%		≥ 107.5	
	4.15	Efficiency at CH Min (4030 condition)	≥ 107.5	%		≥ 107.5	
	4.16	Max. CO_2 Adjustment	-	%		9.1	
	4.17	Min. CO_2 Adjustment	-	%		8.5	
Reliability	5.1	Boiling Limit 0.6 bar / 90ºC	-	%Circ.		No Boiling	Consider pressure sensor to limit lower pressure value
	5.2	Surface Temperatures, including seals	< 120	ºC		< 120	< 160ºC in abnormal working conditions
	5.3	Mechanical Stress causing Deformation	Not allowed	-		Not allowed	
	5.4	Thermoacoustic noise	Not allowed	-		Not allowed	

Legal	6.1	Comfort according EN13203-1	≥ 40	Points	X	42	Applicable to UK only
	6.2	NO_X Class	5	-		5	
	6.3		< 70	mg/kWh		< 70	
	6.4	CO_{DAF} at Max. Rated Heat Input	< 100	ppm		< 100	
	6.5	Metals in Condensate water according ATV-DVWK-A 251 (applicable to DACH only: Germany, Austria and Switzerland) - Lead - Cadmium - Chromium - Copper - Nickel - Zinc - Tin - Sulfates - Nitrites - Nitrates	See Table	mg/l		See Table	- Lead < 0,2 mg/l - Cadmium < 0,01 mg/l - Chromium < 0,2 mg/l - Copper < 0,3 mg/l - Nickel < 0,2 mg/l - Zinc < 0,3 mg/l - Tin < 0,1 mg/l - Sulfate < 40 mg/l - Nitrite < 6 mg/l - Nitrate < 40 mg/l
	6.6	Blau Engel - CO Emissions	< 20	mg/kWh		< 20	Applicable to DE only - Optional Labelling
		Blau Engel - NO_X Emissions	< 40	mg/kWh		< 40	

Figure 19 – Technical Requirements for the heat cell

4.3 Standards to Comply

To be able to be commercialized in current markets where Bosch is present, the new heat cell must comply with all the relevant standards presently active, as well as expected changes or introductions:

European Directives:
Gas Appliance Directive (2009/142/EC)
Boiler Efficiency Directive (92/42/EEC)
Low Voltage Directive (2006/95/EC)
Electromagnetic Compatibility Directive (2004/108/EC)
Drinking Water Directive (98/83/EC)
Eco Design Directive (2009/125/EC), Regulation for space heaters (Lot1)
Labeling Directive (2010/30/EU), Regulation for space heaters (Lot1)

Figure 20 – European directives for gas fired heating appliances not exceeding 70kW

National Laws:
1.BimSchV (DE)
TrinkwV 2001 (DE)
Art. 15a B-VG (Juni 1995) der österreichischen Ländervereinbarung (AT)
Schweizer Luftreinhalteverordnung (CH)
Niederländisches NOx-Gesetz (NL)

Figure 21 – National specific laws for gas fired heating appliances

Marks:
CE
VDE (Gerätesicherheit und EMV) (DE)
ÖVGW (AT)
SVGW (CH)
HR Top (BE)
GASKEUR SV, HR-107, HRww, CW, NZ (NL)

Figure 22 – Existing Marks depending on introduction market

Standards and Test Regulations:
EN 15502, Gas-fired heating boilers
EN 297, Gas Boilers, Type B < 70 kW (replaced by EN 15502)
EN 656, Gas Boilers, Type B, 70 until 300 kW (replaced by EN 15502)
EN 483, Gas Boilers, Type C < 70 kW (replaced by EN 15502)
EN 625, Combi Boilers (replaced by EN 15502)
EN 677, Gas Condensing Boilers < 70 kW (replaced by EN 15502)
EN 13203, Domestic Hot Water, Performance/Energy Consumption
EN 60335-2-102, Electrical Equipment of non-electric appliances
EN 62233, Electromagnetic Fields
EN 55014-1, EN 55014-2
EN 61000-3-2, EN 61000-3-3
Environmental Mark Blauer Engel RAL-UZ 39, Gas-Spezialheizkessel (DE)
Environmental Mark Blauer Engel RAL-UZ 40, Gas-Umlaufwasserheizer (DE)
Environmental Mark Blauer Engel RAL-UZ 61, Gas-Brennwertgeräte (DE)
KTW-Requirements for plastic materials in Drinking water (DE)
DIN 50930-6, Corrosion of metals / Drinking water (DE)
ÖVGW PG 300, PG 307, PG 346, PG 347, PG 357, PG 359 (AT)
GASKEUR Requirements: Basis CV, SV, HR, CW / HRww, NZ (NL)
NF D 30-002, NF D 30-010, NF EN 23741 (FR)
WRAS (GB)
SEDBUK, Band A (GB)

Figure 23 – European directives for gas fired appliances not exceeding 70kW

Promotion Programs:
proKlima, Hannover
Hamburg
SAB (Sachsen)
KfW
BAFA

Figure 24 – Existing promotion programs depending on introduction market

Additionally, it has to comply with specific Bosch directives, robustness and reliability procedures:

Reliability / Robustness Test Procedures – Heat Exchanger:
Thermal Shock Stress with Flow
Thermal Shock Stress without Flow
Flue Gas Corrosion Test
Boiling through Calcification

Reliability Test Procedures – Seals:
Material Requirements for EPDM o-rings used in CH and DHW
Material Requirements for elastomeric seals in contact with flue gas and condensate

Safety Directives:
Safety Directive “Flue Gas Way“”

Figure 25 – Bosch directives, robustness and reliability procedures applicable to heat exchangers

5 Technical Chalenges

5.1 Risk Management Sheet

After the concept creation, the next step is to review the design and assess the potential risks. This will show the potential failures, their consequences and will direct the team in their elimination or minimization.

For each detected potential risk, an impact assessment and probability of occurrence are defined according to the following matrix:

Probability			Impact		
Score	Description		Score	Description	on Product
5	Critical (Risk is certain to occur)		5	Critical (Impact is extreme large)	End products unusable
4	High (Risk is highly likely to occur)		4	High (Impact is large)	Important functions are affected, product can be used only with clear restrictions
3	Medium (Risk is somewhat likely to occur)		3	Medium (Impact is moderate)	Important functions are affected, product can be used only with restrictions
2	Low (Risk is unlikely to occur)		2	Low (Impact is small)	Restrictions of the functionality for end user noticeable, but important funcitons are untouched, product can be used
1	Insignificant (likelihood is insignificant)		1	Insignificant (Impact is insignificant)	Restrictions of the functionality for end user hardly noticeable

Figure 26 – Probability and Impact description for assessing risk value

The Risk Value can be presented in a Risk Portfolio Chart like the one shown below:

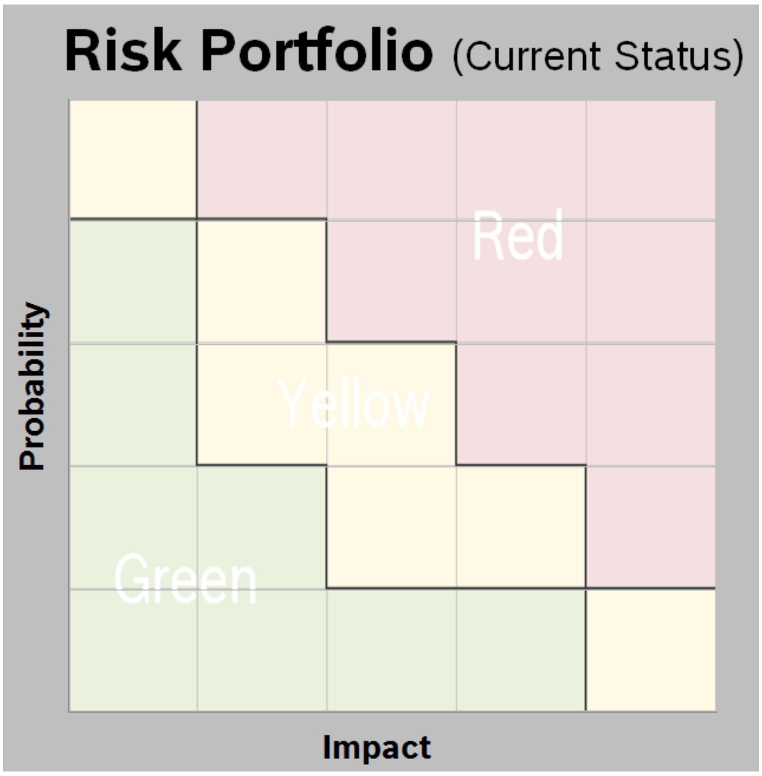

Figure 27 - Risk Porfolio Chart – accepted risks (green), risks requiring risk assessment (yellow) and unacceptable risks (red)

After that, measures are defined to overcome the detected risk, with a responsible person or team and due date. The risk will later be re-evaluated by the team after the completion and a set of documentation of defined measures is prepared.

Risk Management Sheet (Project Category: New, Modification)

Last Update:
Updated by:

Risk Identification					Risk Assessment				Risk Handling					Risk Controlling				
Risk Description	Category	Source	Entry Date	Update Date	Impact Description	Impact	Probability	Risk Index	Response Strategy	Measure	Owner	Due Date	Current Status	Result	Impact (after action)	Probability (after action)	Current Risk Index	Keyword for Portfolio
Air temperature at the fan inlet > 80°C	Product / Technology	Team	5-Mar-13	18-Apr-13	Customer complaint due to no performance: fan failure. Lifetime not met (electrical component failure).	5	2	10	Eliminate	1. Check at worst case conditions (minimum load, with propane, maximum supply temperature, post purge, ...) 2. Design the front module water channels in order to remove hot spots.	Team		In track		5	2	10	Lifetime
Air temperature at the gas valve > 70°C	Product / Technology	Team	5-Mar-13	18-Apr-13	Customer complaint due to no performance: gas valve failure. Lifetime not met (sticking seal failure).	5	2	10	Minimize	1. Check at worst case conditions (minimum load, with propane, maximum supply temperature, post purge, ...) 2. Design the front module water channels in order to remove hot spots.	Team		In track		5	2	10	Lifetime
Blockage of water channels due to calcification.	Product / Technology	Team	5-Mar-13	18-Apr-13	Multiple failures possible: - boiling noises (customer complaint); - cavitation effects on steel: pitting and leakage; - overheating also leading to material failure and leakage; - lower thermal heat transfer performance.	5	2	10	Minimize	1. Perform artificial blockage tests to assess boiling and performance due to calcification. 2. Perform RHE02 scaling test. 3. Check impact of additives.	Team		In track		5	2	10	Lifetime / Safety

Blockage of water channel due to debris from CH system.	Product / Technology	Team	5-Mar-13		Multiple failures possible: - boiling noises (customer complaint); - cavitation effects on steel: pitting and leakage; - overheating also leading to material failure and leakage; - lower thermal heat transfer performance.	5	2	10	Minimize	1. Perform artificial blockage tests to assess boiling and performance due to calcification. 2. Perform RHE02 scaling test. 3. If required add a filter on the CH line of the appliance hydraulic unit	Team		In track		5	2	10	Lifetime / Safety
Blockage of flue gas channel due to condensate trap.	Product / Technology	Team	5-Mar-13	18-Apr-13	Appliance lock-out: customer complaint	4	2	8	Eliminate	1. Measure condensate retention in HE. 2. Measure if drop of power output exists after long operating times.	Team		In track		4	2	8	Market Requirements
Local zones within water channels which enables air to be trapped (local dead zones, recirculation zones, etc.)	Product / Technology	Team	5-Mar-13	18-Apr-13	Local boiling, leading to increased corrosion and erosion leakage. Customer impact also regarding noise.	5	1	5	Eliminate	1. Check severity of aeration. 2. Improve design if necessary to improve de-aeration.	Team		In track		5	1	5	Lifetime / Safety / Standard Compliance
The gap between modules for gas passages is out of tolerance.	Product / Technology	Team	5-Mar-13	18-Apr-13	Reduced efficiency for flue gas side if the gap is too wide; Drop in power and increased risk for blockage if the gap is too narrow.	2	2	4	Eliminate	1. Check process capability with Cpk and Cpm values.	Team		In track		2	2	4	Standard Compliance
CH efficiency not fulfilling requirements.	Product / Technology	Team	5-Mar-13	18-Apr-13	Non-conformity of appliance or higher line rejects	5	1	5	Eliminate	1. Check how evenly distributed the exhaust flow is and measure impact on efficiency; 2. Measure variation in localised flue gas temperature using "temperature flue gas scan" setup.	Team		In track		5	1	5	Market Requirements

Non-uniform combustion on burner or bad flow distribuion at the burner surface, leading to poor emissions, efficiency and thermoacoustics.	Product / Technology	Team	5-Mar-13	18-Apr-13	Non-conformity of appliance or higher line rejects	4	2	8	Eliminate	1. Test performance of combustion. 2. Conduct CFD of flow in the burner. 3. If needed, redesign of gas-air assembly: Repositioning of fan. 4. Possible redesign of burner.	Team		In track		4	2	8	Standard Compliance
Thermoacoustic resonance.	Product / Technology	Team	5-Mar-13	18-Apr-13	Customer complaint due to loud noise and vibration.	4	2	8	Eliminate	Perform thermoaccoustic complete testing.	Team		In track		4	2	8	Market Requirements
High temperatures at ignition assembly seal leading to combustion gases leakage and critical incident.	Product / Technology	Team	5-Mar-13	18-Apr-13	Reduced seal lifetime: critical failure	5	5	25	Eliminate	1. Design the water channels of the front module to achieve better cooling by simulation trials. 2. Check combustion chamber temperatures under blocked fin pack conditions. 3. Execute fail safe tests 4. Calculate the achiavable lifetime with defined user-profile and collective loads	Team		In track		5	5	25	Lifetime / Safety

High temperatures at burner gasket leading to combustion gases leakage and critical incident.	Product / Technology	Team	5-Mar-13		Reduced seal lifetime: critical failure	5	5	25	Eliminate	1. Design the water channels of the front module to achieve better cooling by simulation trials. 2. Check combustion chamber temperatures under blocked fin pack conditions. 3. Execute fail safe tests 4. Calculate the achiavable lifetime with defined user-profile and collective loads	Team		In track		5	5	25	Lifetime / Safety
Stiffness of burner housing leading to burner gasket leakage.	Product / Technology	Team	5-Mar-13	18-Apr-13	Deformation of burner housing leads to critical gas-air or flame leakage.	5	2	10	Eliminate	1. Burner plate design to achieve required stiffness. 2. Perform RHE03 test: thermal crack without flow.	Team		In track		5	2	10	Lifetime / Safety
Corrosion of stainless steel plates on flue gas side.	Product / Technology	Team	5-Mar-13	18-Apr-13	Flue blockage leading to power drop: HE is not usable. Insufficient heat delivery over lifetime. Potential thermal acoustic resonances. Heat comfort is compromised.	4	2	8	Minimize	1. Test and establish consequences. Perform RHE01.A and RHE02.B. 2. If necessary, define software protocol for self-cleaning algortithm variant. 3. If necessary, investigate coating as preventive measure.	Team		In track		4	2	8	Lifetime
Water and flue gas side leakage due to copper brazing failure.	Product / Technology	Team	5-Mar-13	18-Apr-13	Brazing failure due to cyclic thermal shocks. If flue or water side leakage occurs then appliance is no longer functional.	5	3	15	Eliminate	1. Perform thermal shock endurance test. LHE01	Team		In track		5	3	15	Lifetime / Safety

Uniformity of protective coating.	Product / Technology	Team	5-Mar-13	18-Apr-13	Corrosive blocking or cracking of flue passage/plates respectively.	5	2	10	Eliminate	1. Test process capability of supplier; 2. Investigate best coating method OR Switch to Ni brazing	Team		In track		5	2	10	Lifetime / Safety
Exceed heavy metal content limit in condensation.	Product / Technology	Team	5-Mar-13	18-Apr-13	Environmental impact: exceed legal limit for heavy metals in condensate. Cannot release appliance in certain countries (DACH)	5	3	15	Eliminate	1. Operate boiler under critical case condensing conditions (maximum, minimum and nominal loads) and measure condensate heavy metal content. (ref.ATV251).	Team		In track		5	3	15	Standard Compliance
CMS capability not reached	Product / Technology	Team	5-Mar-13	18-Apr-13	not "CMS ready" (unable to fulfill market requirements)	4	2	8	Eliminate	1. Check burner compatibility with CMS 2nd generation. 2. Define potential additional development work needed. 3. Clarifiy with fan, gas valve and burner suppliers this requirement in MOS' clearly.	Team		In track		4	2	8	Market Requirements

Figure 28 – Risk Management Sheet for the selected design concept

The summary of initial status is presented the Risk Portfolio Chart below:

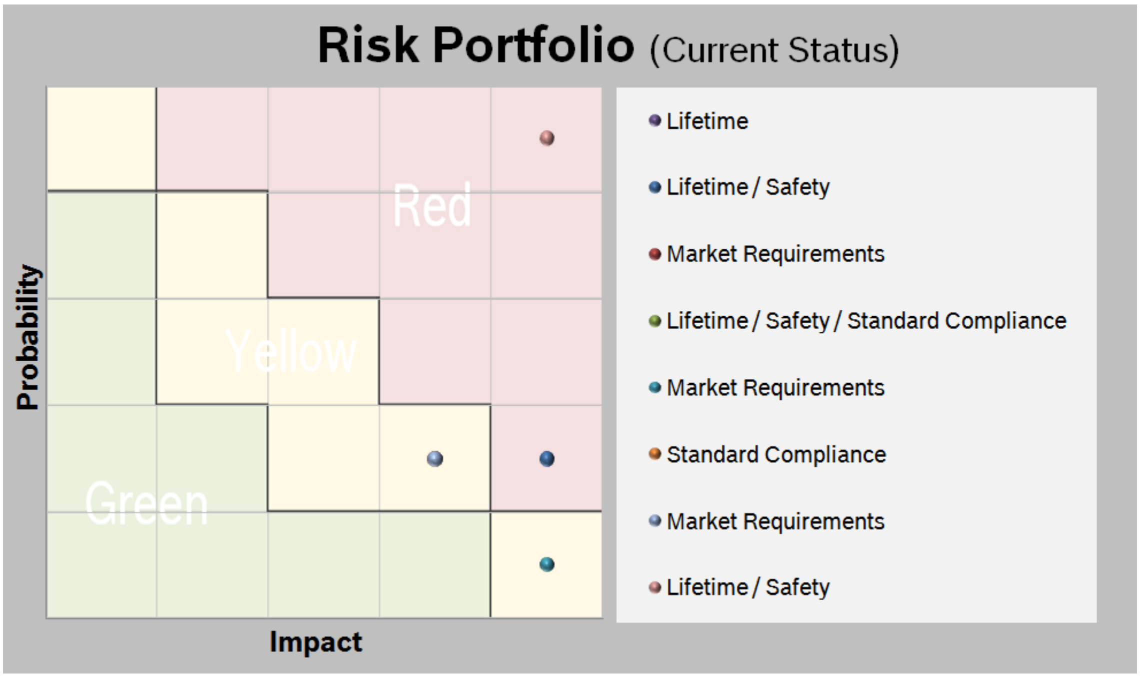

Figure 29 - Initial Risk Portfolio Chart

5.2 Risk Evaluation Methods

The risks presented in Chapter 5, Section 5.1, can be generally presented as:

- Environmental (corrosion resistance; metal content in condensates; temperatures of components);
- Mechanical Strength (heat exchanger stiffness);
- Performance (pollutants emissions; efficiency; thermo-acoustics behavior);
- Lifetime (seals temperatures; flue gas or water circuits' blockage).

The present dissertation will focus on the thermal performance and lifetime, demonstrating in Chapter 6 – Computational Fluid Dynamics, the design features introduced and optimized to maximize efficiency and minimize temperatures in heat exchanger and in critical seals.

In Chapter 7, the laboratory experiments that validate the theoretical analysis will be presented. The laboratory results focus will be given to modulation range, pollutants emissions, efficiency and temperatures of components.

6 Computational Fluid Dynamics

6.1 Procedure

The present CFD study is dedicated to the development, design and improvement of the plate to plate condensing heat exchanger configuration described in Chapter 4.

CFD and heat transfer simulations are performed via ANSYS 13.0 in order to determine thermal performance, to visualize temperature and pressure distribution, and flow structure through a single PHE module.

While hot gas passes through the tight gaps between the plate modules, thermal energy from the gas to the water is maintained on a wide heat transfer surface with a high thermal efficiency via both convection and condensation. Therefore, the design parameters of the plates, i.e. fishbone structure, channel geometry, surface area and thickness, become significant properties which should be carefully considered during their development stage.

Based on the computational results, the plate configuration is progressively redesigned and improved in terms of increasing thermal and condensation efficiency, preventing boiling, and reducing manufacturing cost.

In this study, flow structure and heat transfer characteristics on a number of four different plates are investigated in terms of contours of temperature and pressure, velocity magnitude, and patterns of streamline. All simulations are performed for a 36 kW PHE type condensing heat cell.

ANSYS 13.0 and Fluent were employed for the present CFD analysis. The simulation process was performed by:

- Using ANSYS Design Modeler 13.0;
- Gas and water parts were prepared from CAD models for quality meshing process;
- The model was meshed with MESHING 13.0;
- The model was solved with Fluent via parallel processing (a total of eight processors were employed);
- Analyzed data was post-processed with CFD-post;
- Images and figures shown herein were prepared with CorelDRAW X5.

Modeling:

In order to reduce the processing load and time, CFD and heat transfer analysis were performed for a single PHE module under the symmetry conditions as shown in Figure 30. The module includes a water domain subtracted from the PHE and two half volume gas domains on each side of it. Conduction due to the thickness of the plate was simulated as shell conduction.

Meshing:

Due to the complexity of its geometry, the model was meshed with patch confirming triangular / tetrahedral mesh generation with applying sweeping method on appropriate parts. Then, the mesh elements with high skewness were converted to polyhedral type for faster convergence and processing.

Solution:

Coupled pressure-based solver was employed with species transport method. Energy/realizable k-ε turbulence model was used, and second-order upwind discretization was applied.

Properties of Gas Mixture:

CO_2: 7.93% (v/v);

H_2O: 15.85% (v/v);

O_2: 3.49% (v/v);

N_2: 72.73% (v/v);

Density ρ: incompressible ideal gas;

Specific heat capacity c_P: mixing law;

Dynamic viscosity μ: ideal gas mixing law;

Thermal conductivity k: ideal gas mixing law.

At gas inlet, mass-flow-inlet condition was employed with:

Mass flow rate per module: 0.64 g/s;

Temperature $T_{GAS\ IN}$: 2023 K;

Turbulence Intensity I: 5%;

Hydraulic diameter d_H: 9.63 mm.

At gas outlet, pressure-outlet condition was employed with:

Average Pressure, P_{GAUGE} = 0 bar;

Backflow temperature T_{OUT}: 300 K;

Turbulence Intensity I: 5 %;

Hydraulic diameter d_H: 9.72 mm.

At water inlet, mass-flow-inlet condition was employed with:

Mass flow rate per module: 16.7 g/s;

Temperature $T_{WATER\ IN}$: 333 K;

Turbulence Intensity I: 5%;

Hydraulic diameter d_H: 17 mm.

At water outlet, pressure-outlet condition was employed with,

Average Pressure, $P_{GAUGE} = 0$ bar;

Backflow temperature T_{OUT}: 300 K;

Turbulence Intensity I: 5 %;

Hydraulic diameter d_H: 17 mm.

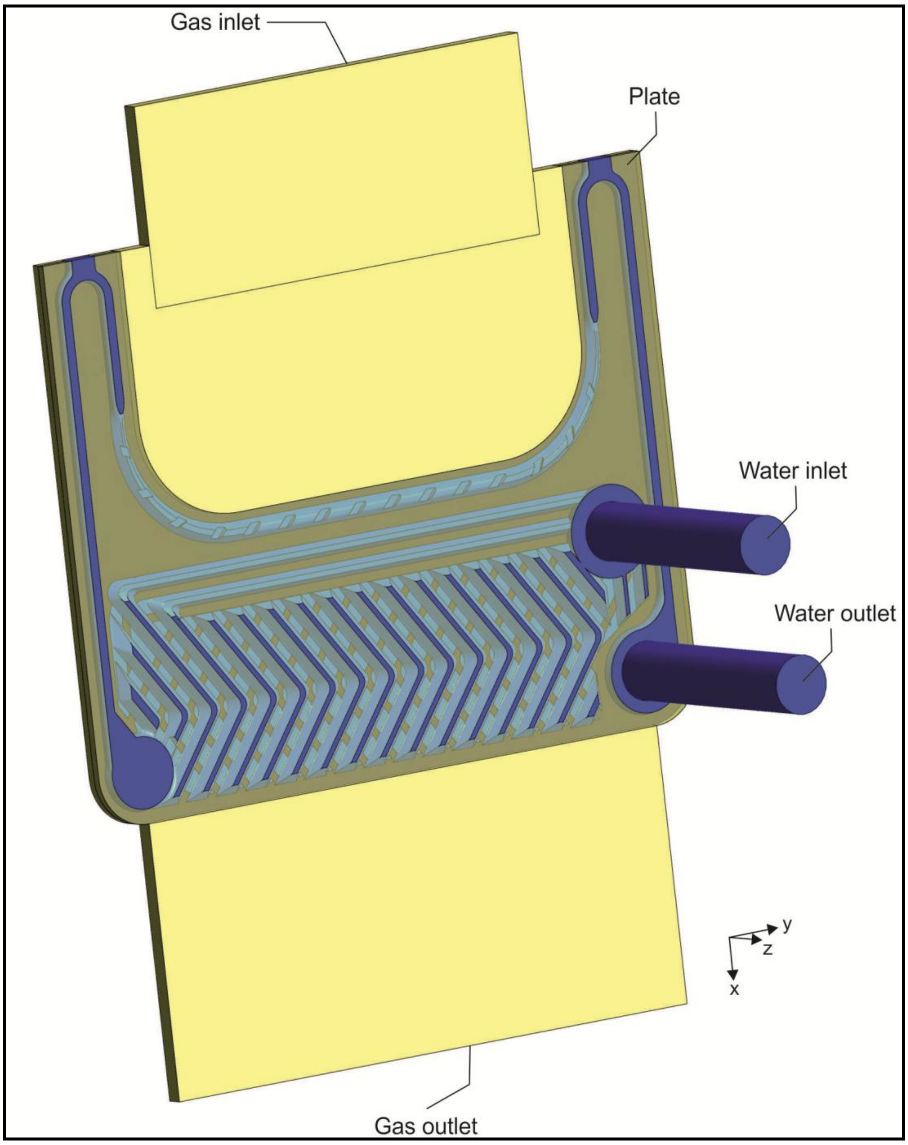

Figure 30 - Perspective view of the domains of Model 4

6.2 Results

All simulations studied herein were performed for 36 KW power input and a number of 25 plate modules. The results are presented with an order which follows PHE's development stages. Four versions of plate configurations, shown in Figure 31, were designed to be used in PHE type heat cell.

The first and second models have a rectangular plate configuration and represent concept 1 presented in Chapter 4. Two by-passes were introduced into Model 2 in order to prevent boiling problem in the first two top channels by letting the cooler water to pass through into these channels.

Model 3 and 4 are designed with a U-shaped geometry which eliminates using of aluminum casing with water cooling on the top of the heat cell. Therefore, manufacturing cost can be reduced and leak proofing can be provided. In addition, small punches with an angle directing the water towards to the top edge are located on the first top channel. The aim is that higher amount of water will be forced to pass through the top edge where the boiling risk is the highest.

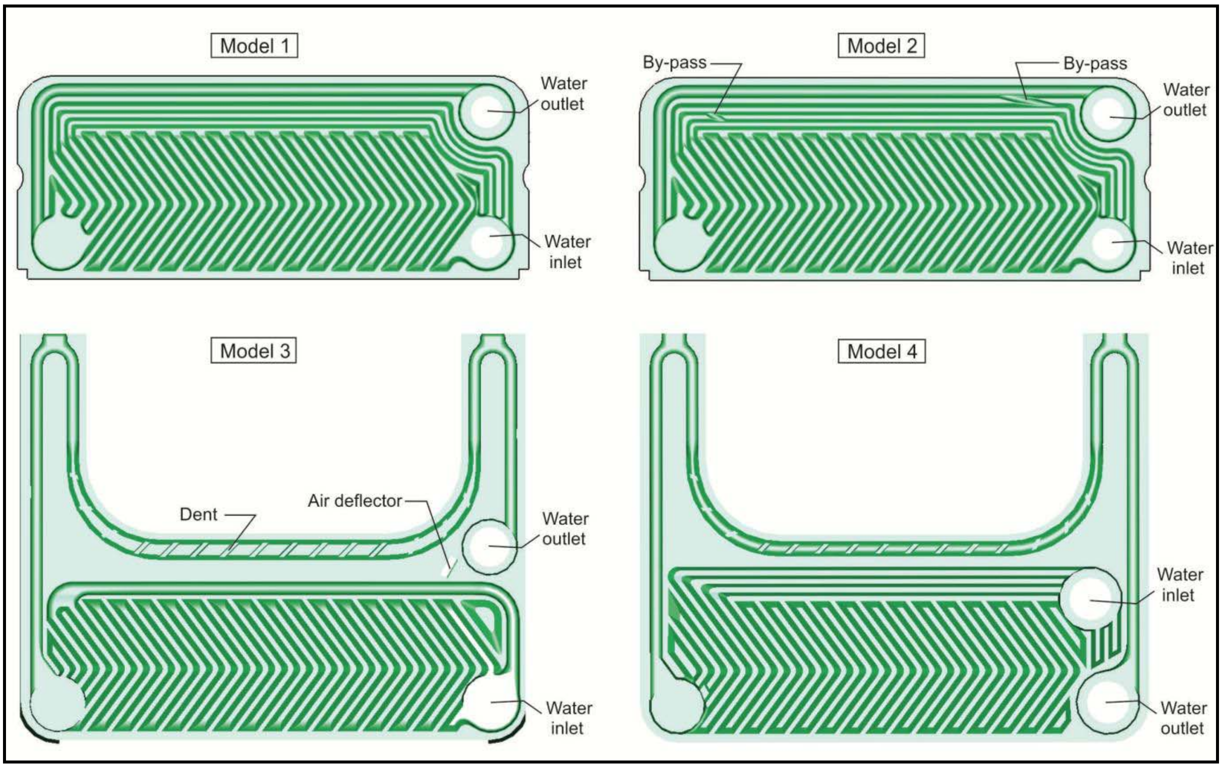

Figure 31 – Plate Configurations analyzed in simulations

Figure 32 shows the contours of temperature, pressure and velocity magnitude distribution on Model 1.

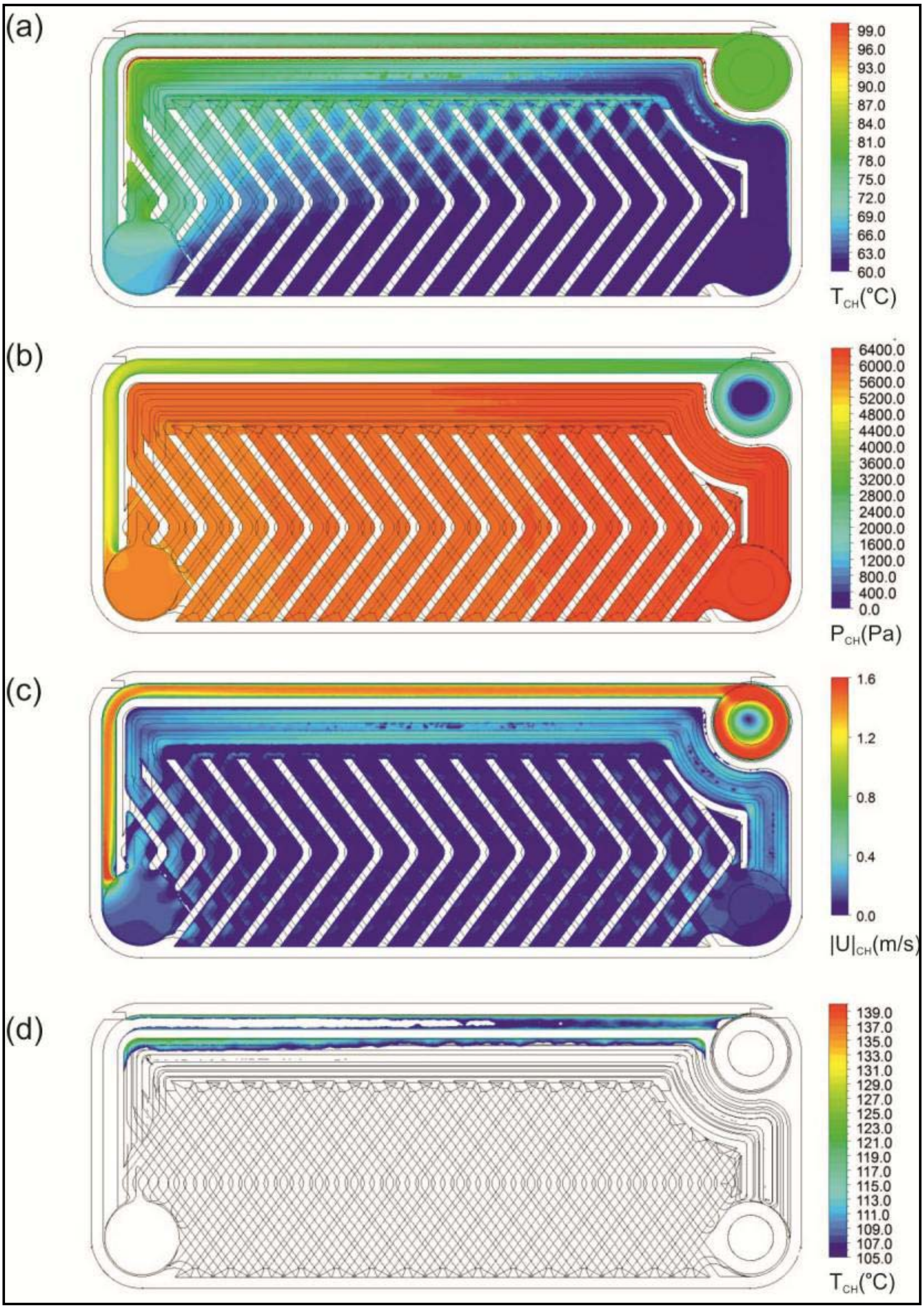

Figure 32 - (a) Contours of temperature; (b) contours of pressure; (c) contours of velocity magnitude on the symmetry plane; (d) contours of temperature on the water surface of Model 1

The top image in Figure 32 (a) represents the temperature distribution in the range of 60 ºC to 100 ºC of the water on the symmetry plane. The results showed that the CH water is heated up to 80.7 ºC with this configuration. In terms of the thermal performance, Model 1 provides required heat transfer with expected efficiency from hot gas to the water. However, temperature reaches up to 100 ºC and higher in some regions of the first and second top channels as seen in Figure 32 (a). Figure 32 (d) shows the regions where the water surface temperature is higher than 105 ºC, which creates a potential risk for boiling noises. In order to eliminate those regions, two by-passes from lower channels to top channels are introduced into Model 2.

The total pressure drop of CH water through the plate is calculated as 6305 Pa, and the pressure distribution is shown in Figure 32 (b). The pressure contours indicate that the higher pressure drop appears in the vicinity where the sharp turns (90º elbows) are evident. Figure 32 (c) shows the contours of velocity magnitude.

The streamline patterns superimposed with the temperature contours of the gas domain are presented in Figure 33. On the right side of the figure, it is shown the profile of the temperature drop along the plate.

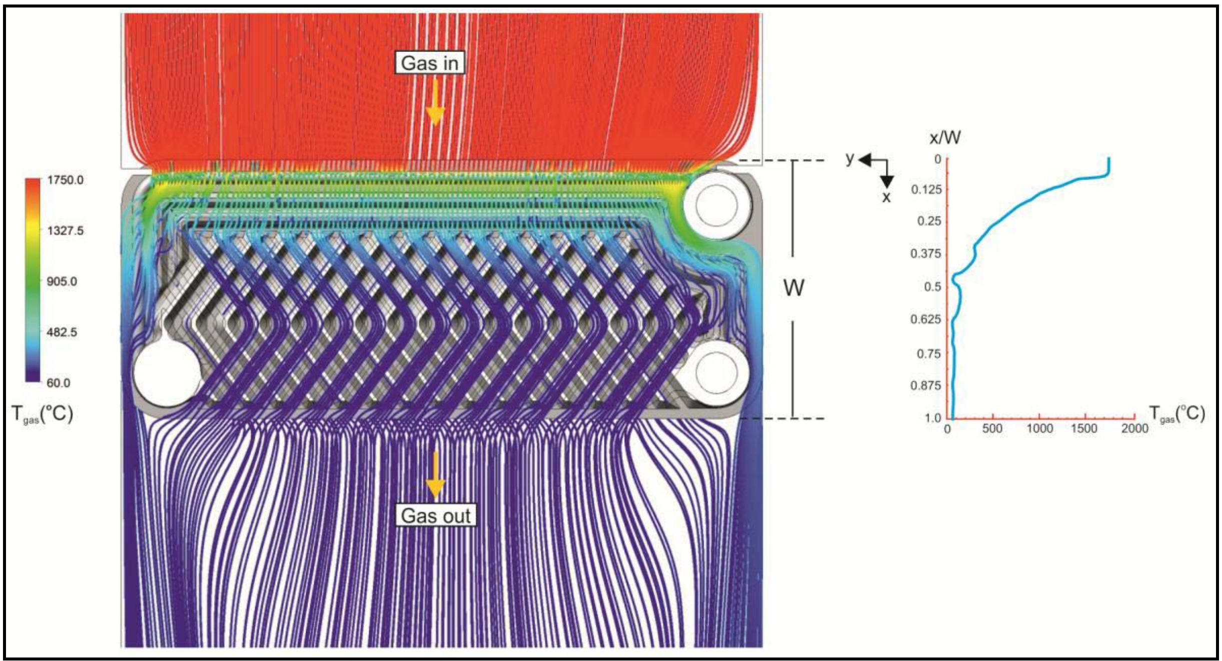

Figure 33 - Streamline patterns of gas domain superposed with temperature on Model 1

It is evident that the temperature of the gas exponentially decreases from 1750 ºC to lower than 100 ºC over the top half of the plate where x/W is between 0 and 0.5. Therefore, significant amount of water vapor inside the flue gas can condense over the second half of the plate after a rapid heat transfer on the top of the plate. On the other side, this sudden cooling will probably have a negative effect on CO emissions.

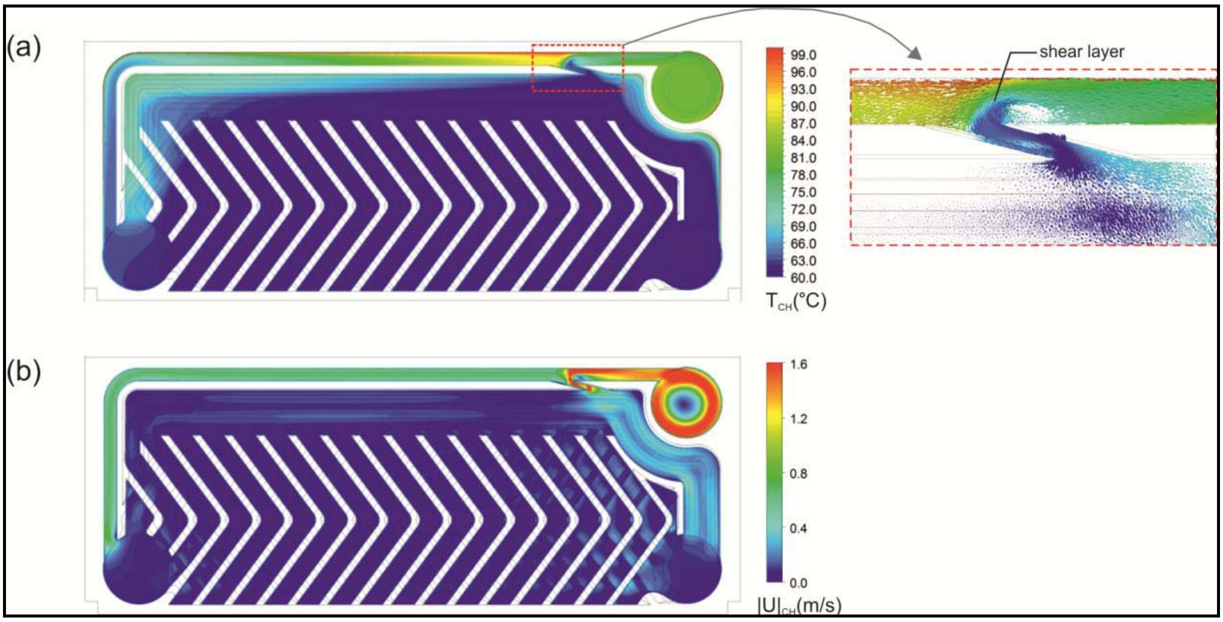

Figure 34 - (a) Contours of temperature; (b) contours of velocity magnitude on the symmetry plane of Model 2

Figure 34 shows the contours of temperature and velocity magnitude on the symmetry plane of Model 2. As described previously, two by-pass channels are integrated into the Model 2 in order to cool down the water temperature inside the top channel. But, the results indicate that the flow passing through the by-pass channel encounters with the incoming water, and they behave like a barrier against each other because of the separation of the flow at the exit corner of the by-pass channel, and the formation of a large scale vortex structure. In the zoomed-in image, the shear layer and the vortex formation are shown with three-dimensional velocity vectors which are colored with the temperature contours. Due to this phenomenon, mass flow rate and flow velocity inside the top channel are reduced at the upstream of the by-pass region which yields the appearance of large regions carrying higher boiling risk.

In order to eliminate the boiling problem inside the top channel, six versions of channel configurations with different geometries were developed, and investigated via CFD and heat transfer simulations. The results are presented in Figure 35. The results show that the highest temperature appears on the tip region of the plate where a large amount of thermal energy is stored. Therefore, water temperature gets higher close to those regions, which can cause the water to boil.

Cross-comparing the images in Figure 35, it is evident that the water temperature close to the surface of the channel on the models with elliptical channel configuration is lower compared to the temperature on the models with the circular geometry. It can be concluded that elliptical geometry is a more promising configuration to prevent boiling inside the top channel of PHE for the aforementioned application. In addition to the geometric improvement, when a thin flat plate is integrated inside the top channel of the module, as applied on Versions 2 and 4, the temperature of the water at the surface and the tip regions can be reduced considerably. As shown in second and fourth row of images in Figure 35, a flat plate provides more thermal energy to be released to the water from the tip region of the plate. Thus, the boiling of the water can be prevented with those plate configurations. But, this type integration during the welding process of the plates makes its manufacturability more challenging and difficult.

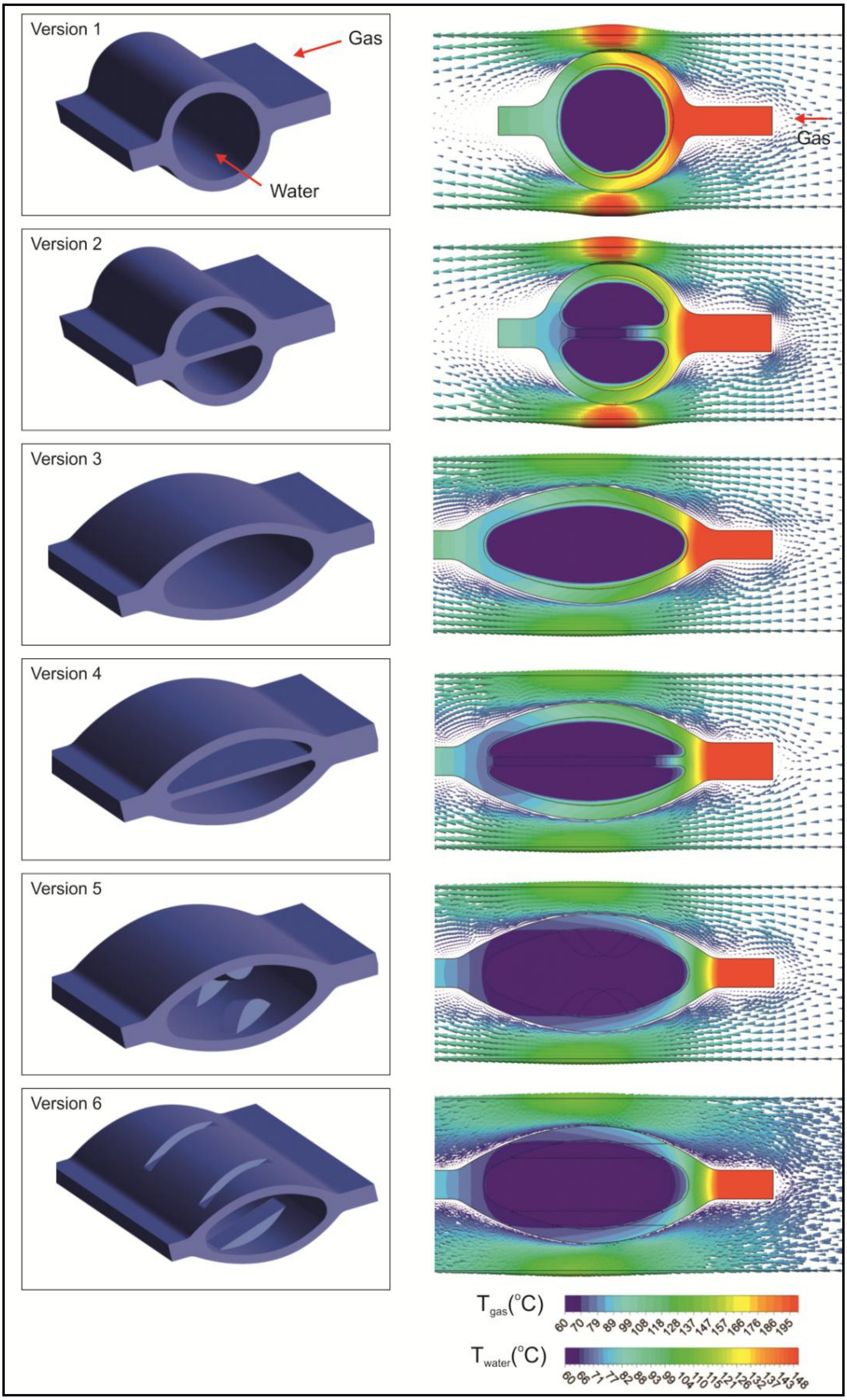

Figure 35 - Proposed top channel configurations for preventing boiling

Another method proposed herein to prevent the boiling is to use dent patterns punched on the surface of the top channel which directs the water flow towards to the high temperature front tip regions.

Two types of dent configuration are applied on Version 5 and 6, as shown in the fifth and sixth rows of images in Figure 35. While the dents force more water to pass through closer to the tip, they also provide more thermal energy to be transferred from the plate to the water by increasing the turbulence. Thus, the highest water temperature can be lowered below 100 ºC. Considering its manufacturability and implementation, Version 6 can be chosen as a convenient preliminary design in this application.

The top channel has been designed with the aforementioned elliptical geometry and implemented on Model 3 and 4 as shown in Figure 31. There are two main differences between the configurations of Model 3 and 4. While Model 4 has a total of three straight top channels, Model 3 has only two. One top channel is removed on Model 3 in order to reduce CO emissions. The other difference is the location of the water inlet and outlets. Water inlet is located in upper part of Model 4 in order to prevent hot gas to escape through the sides.

Both models have a U-shaped configuration which eliminates the use of an aluminum casing. Thus, the heat cell becomes a complete single piece. An air deflector is introduced onto Model 3 in order to direct the hot gas towards to the plate surface. Figures 36 and 37 show the CFD and heat transfer simulation results of Model 3 and 4 respectively.

The top left images in Figure 36 and 37 represent the contours of temperature of the water between 60 ºC to 100 ºC on the symmetry plane. A zoomed-in image of a cross-sectional plane on the top channel is also shown to visualize the critical high temperature regions of the water which can have the boiling problem. In addition to the temperature distribution, contours of pressure and velocity magnitude on the symmetry plane are presented in the bottom left and top right images of Figure 36 and 37 respectively. Finally, the streamline patterns of the gas superposed with the temperature is shown in the bottom right image.

A cross-comparison of the simulation results of Model 3 and Model 4, provides the followings conclusions:

- Model 4 has lower water temperature distribution at the tip region of the top channel as shown in the zoomed-in images, which makes it more convenient model for preventing boiling;
- The temperature over the plate of Model 4 is lower compared to Model 3 at the bottom region where condensation is expected. Thus, the amount of thermal energy absorbed from condensation can be higher on Model 4;
- The modified channel configuration and revised locations of the water inlet and outlet on Model 4 successfully prevent hot gas to escape from the sides;
- While total pressure drop on the water side is 7660 Pa for Model 3, it has been found to be 9970 Pa for Model 4;
- Water temperature at the outlet is 80.59 ºC and 81.23 ºC for Models 3 and 4 respectively.

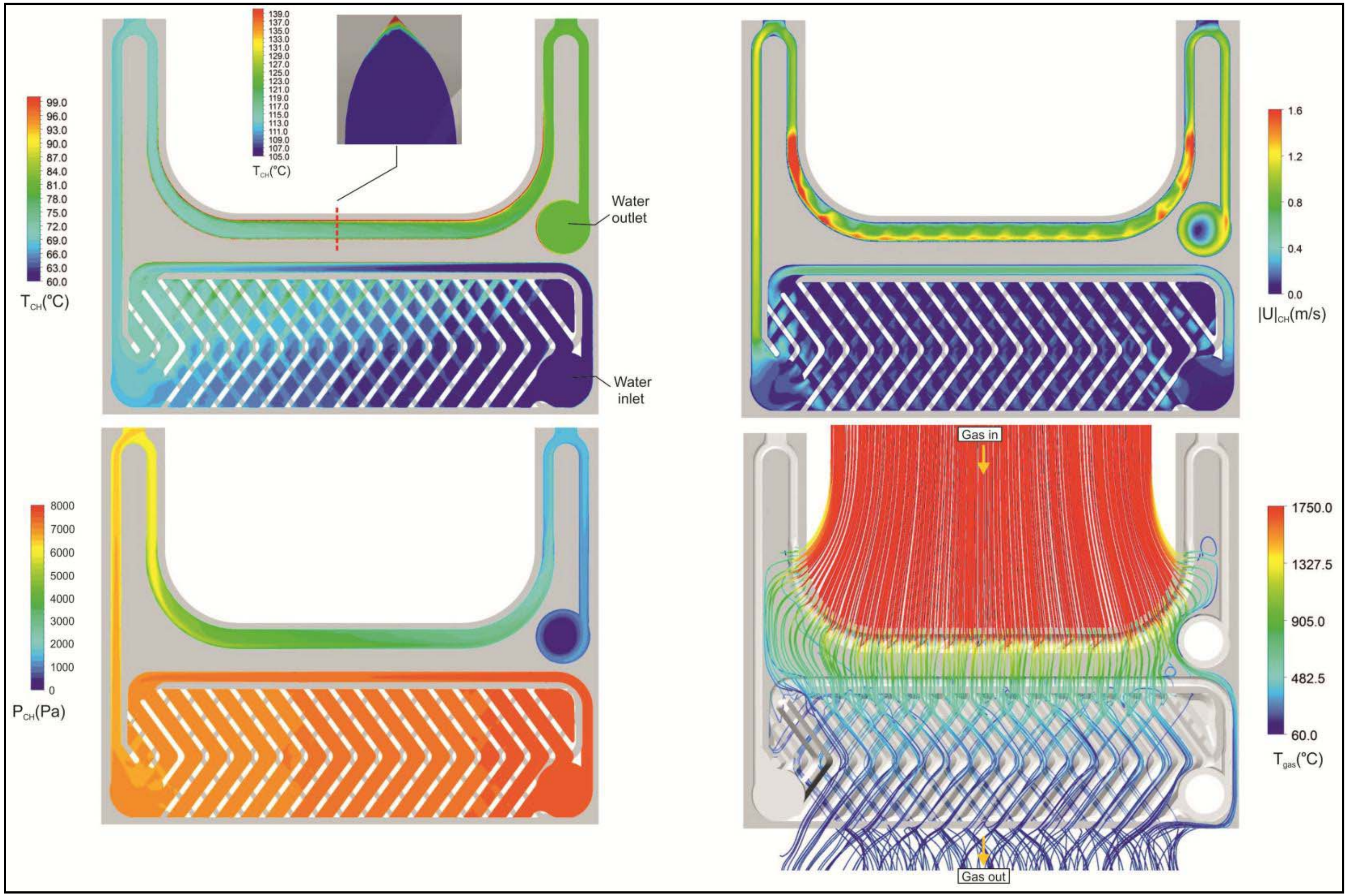

Figure 36 - Patterns of temperature contours (top left), pressure contours (bottom left), velocity magnitude (top right) of the water domain on the symmetry plane, and patterns of streamline (bottom right) of gas domain for Model 3.

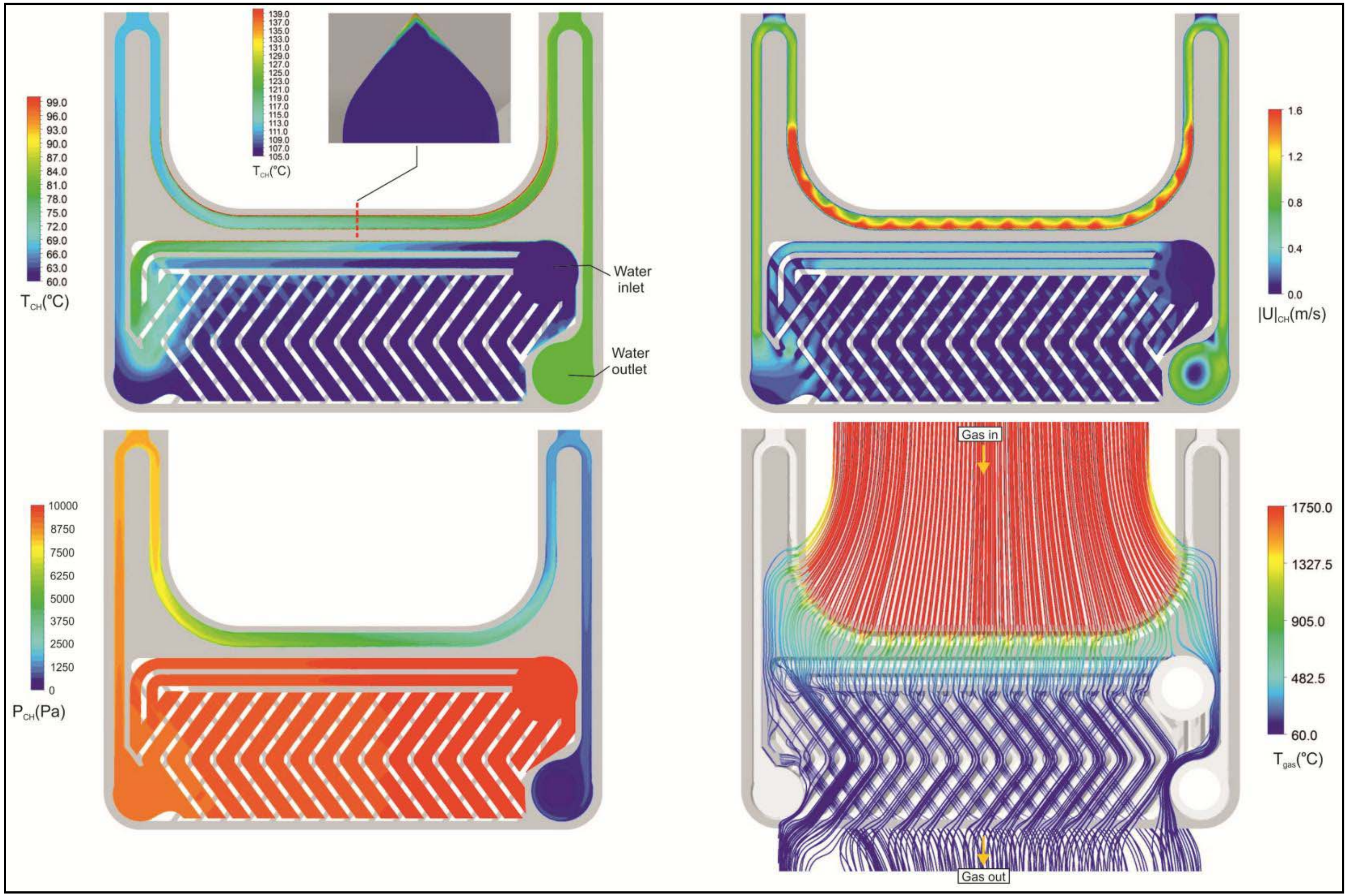

Figure 37 - Patterns of temperature contours (top left), pressure contours (bottom left), velocity magnitude (top right) of the water domain on the symmetry plane, and patterns of streamline (bottom right) of gas domain for Model 4

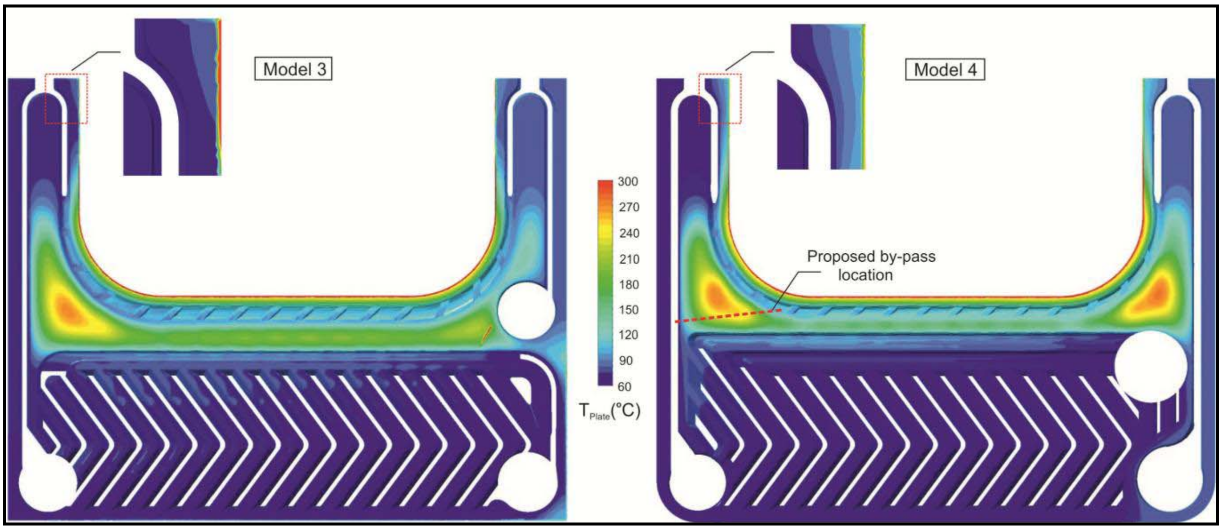

Figure 38 - Temperature distribution on the plate surface

As an overall conclusion, considering the aforementioned application, Model 4 has been found as the most convenient preliminary design among the models studied herein. Although, it provides the required thermal efficiency, the pressure drop on the water domain should be optimized. Moreover, high temperature regions on the plate surface should be minimized as much as possible for the maximum efficiency. Both can be improved by integrating a by-pass channel as proposed in Figure 38. The size and the location of the by-pass channel should be very carefully examined for a proper design.

Alternatively, a hole can be placed in the same location. By doing that, it is expected that the hot spot in the material is eliminated and stresses are reduced.

The need for any of the above design features will be decided after experimental evaluation.

The same assessment was conducted for the front module where there is a critical area where cooling possibility is reduced: igniter assembly location. The igniters have to be placed close to the burner surface in order to safely ignite the gas / air mixture. This is, however, where the flue gases have the highest temperature.

The igniter assembly requires a flat surface to seal the flue gas. This flat area implies that cooling is highly reduced in this region (see Figure 39).

Furthermore, there is also conduction through the igniter pins that will probably heat the external surface of the seal to a value higher than its specification (see Figure 40).

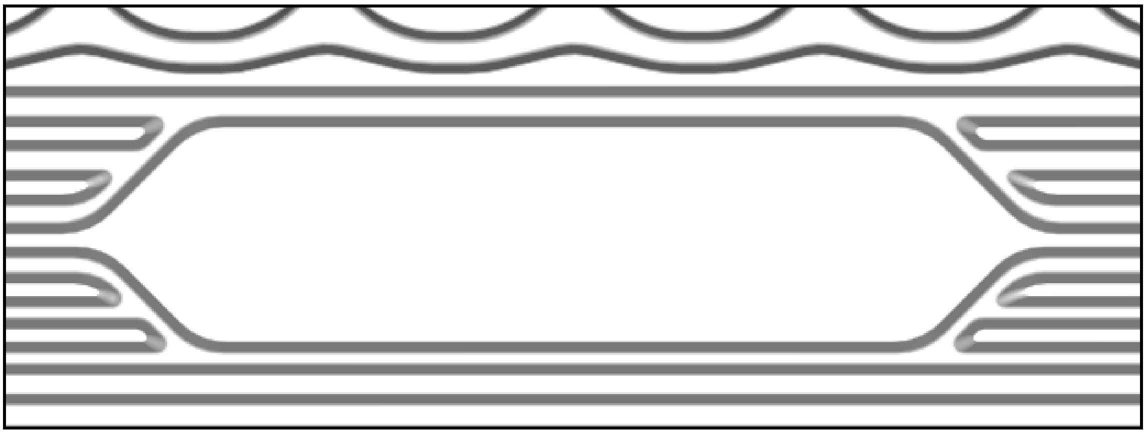

Figure 39 – Flat surface required for igniter sealing

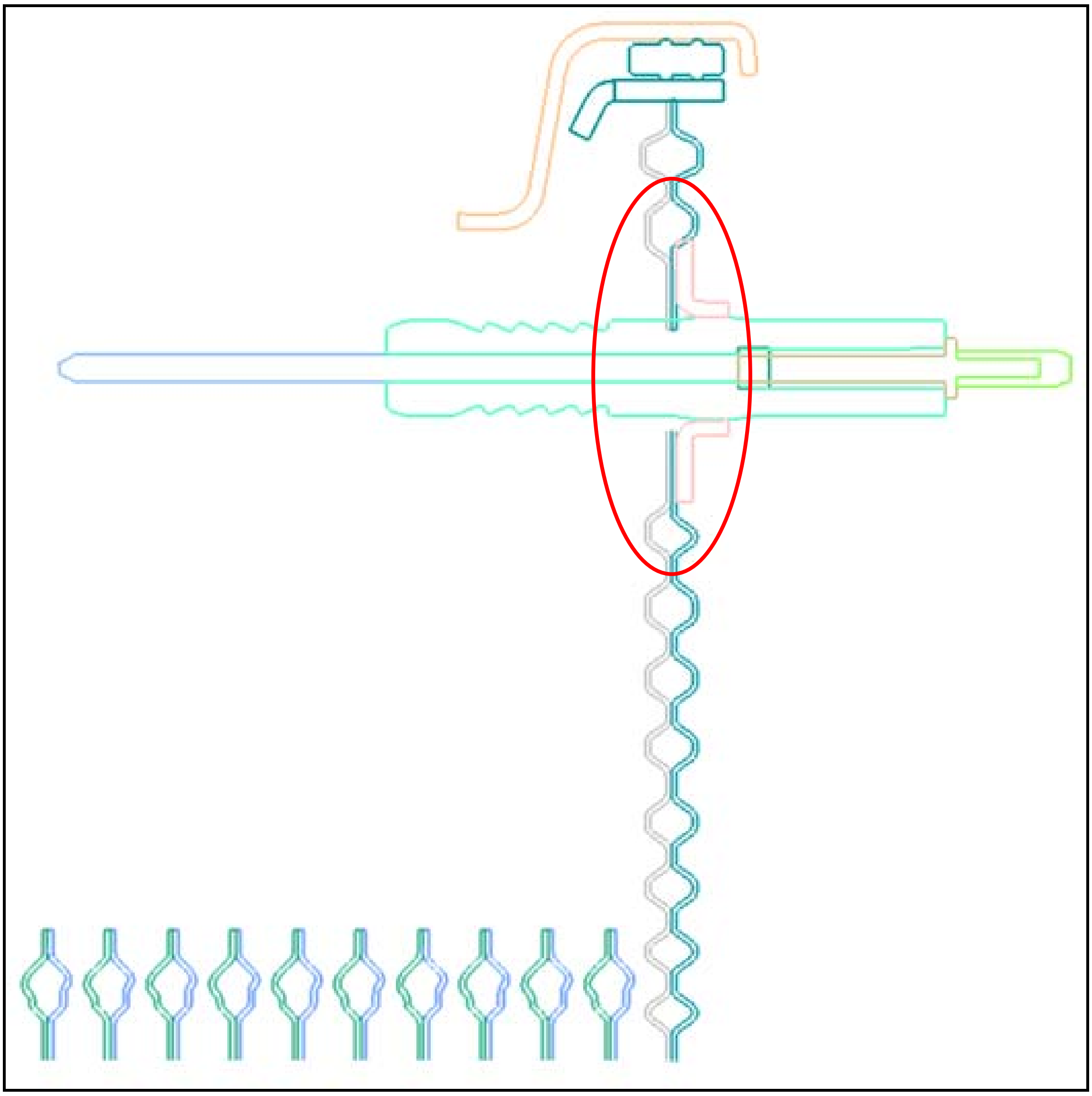

Figure 40 – Assembly of the igniter in the front module of the heat exchanger

For the front module simulation, each module is represented by a solid sheet in order to realistically direct the gas mixture towards the end plate. Symmetry conditions are applied to the system and only 13 channels are included in the model.

The water flow through the middle plates is underestimated and the proportion of gas/water volume is kept as before. Flow channels are taken into account in the end plate.

Ambient is defined next to end plate with 40 °C wall temperatures, to simulate real conditions found inside the boiler casing. Thickness of end plate is defined as 1 mm and shell conduction function is used for heat transfer calculations.

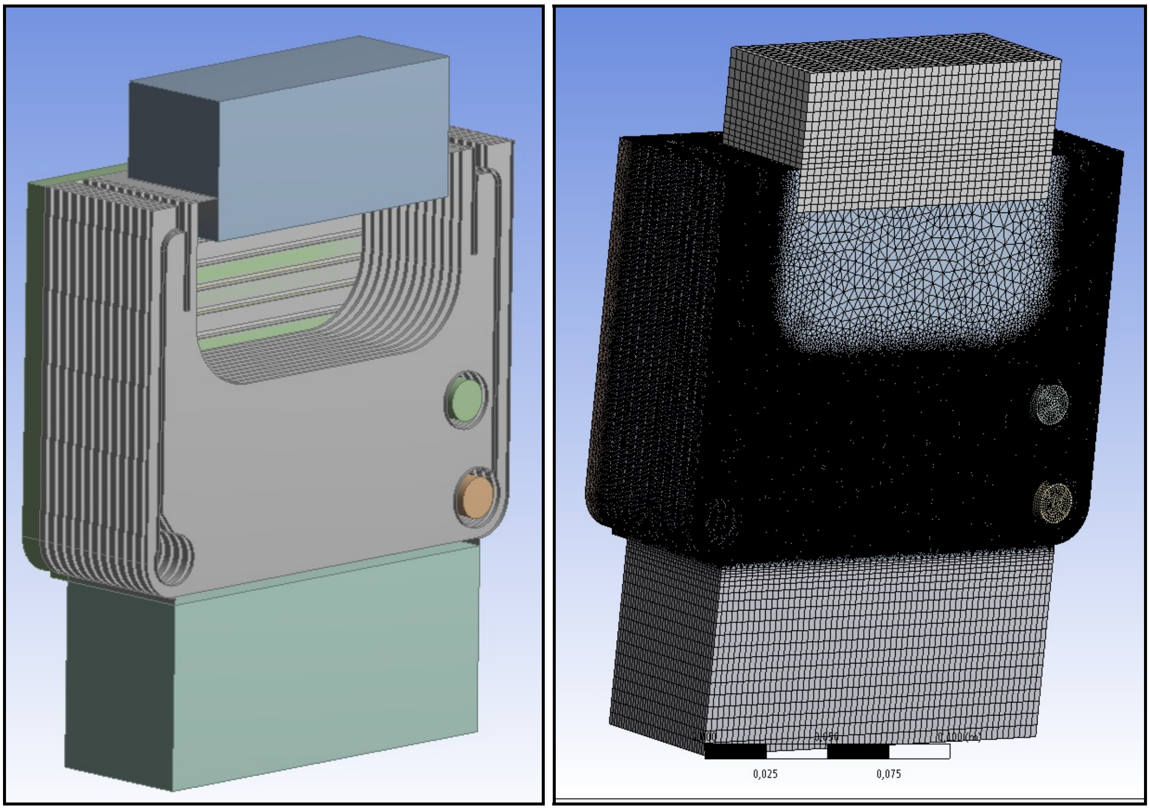

Figure 41 - Perspective view of the domains used and applied 3.6 millions mesh

In Figures 42, 43, 44 and 45 the contours of surface temperature, water temperature, pressure and velocity respectively of the external front module can be seen.

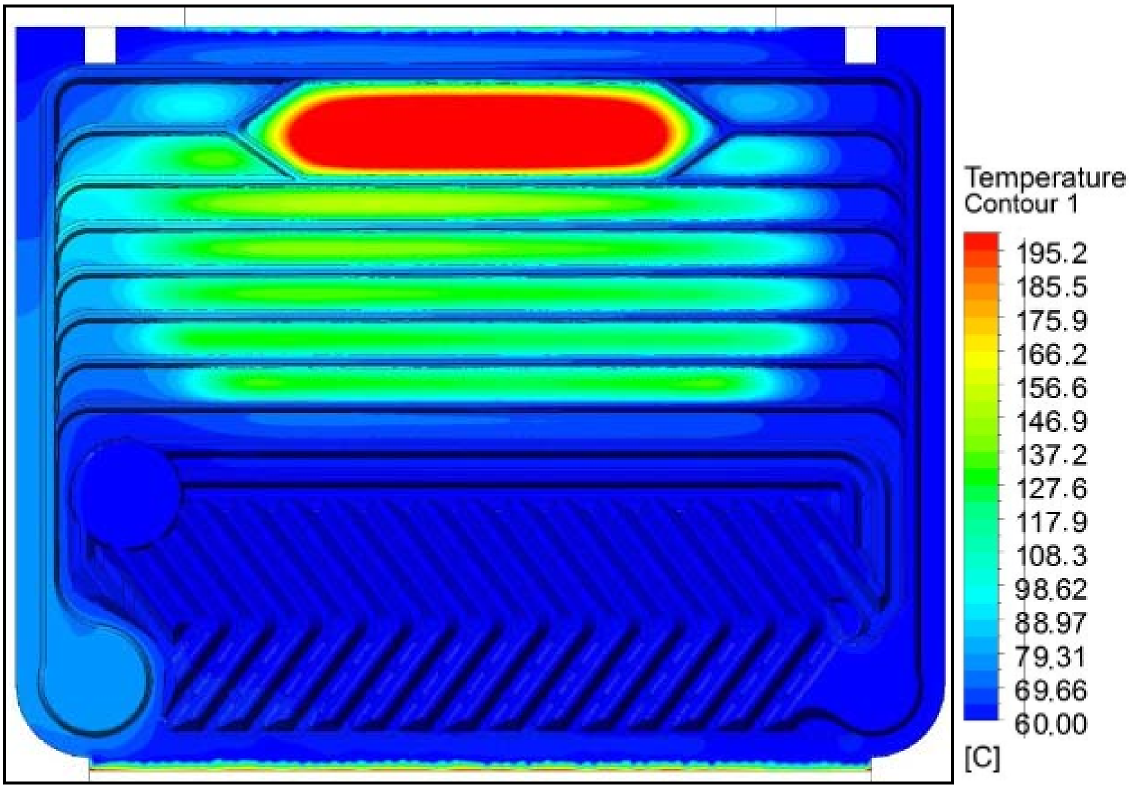

Figure 42 – Temperature contour of the external front plate

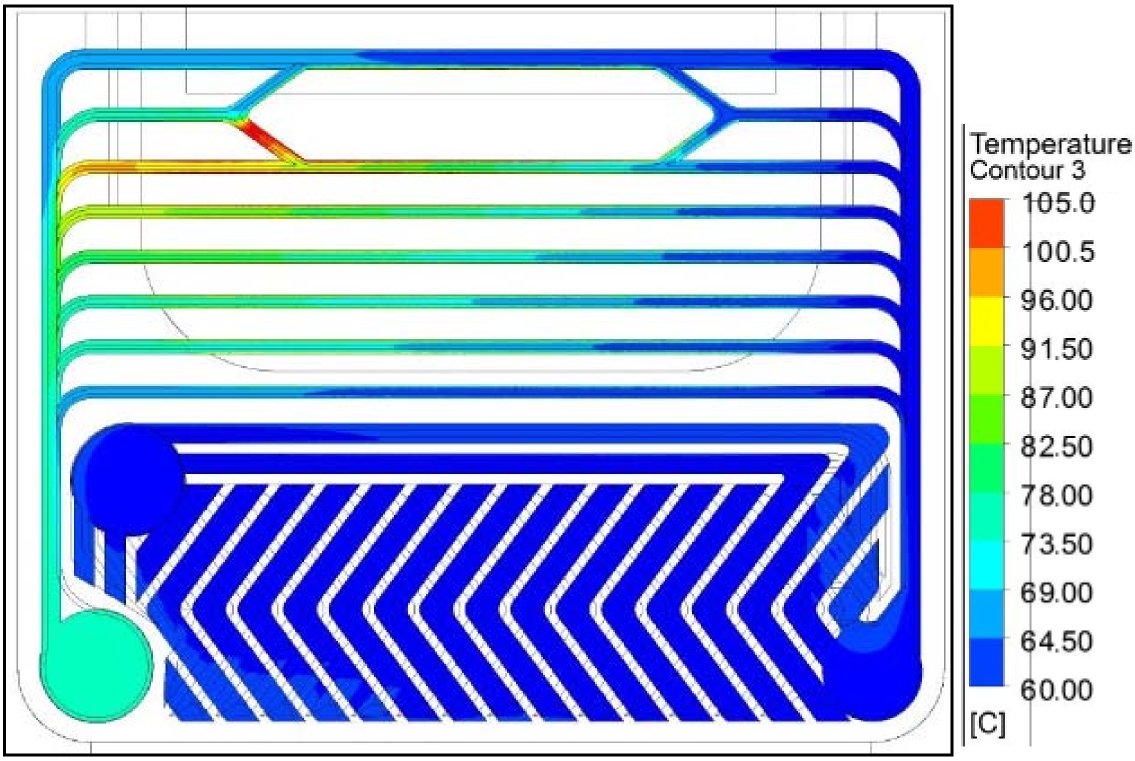

Figure 43 – Water temperature contour in the external front module symmetry plane

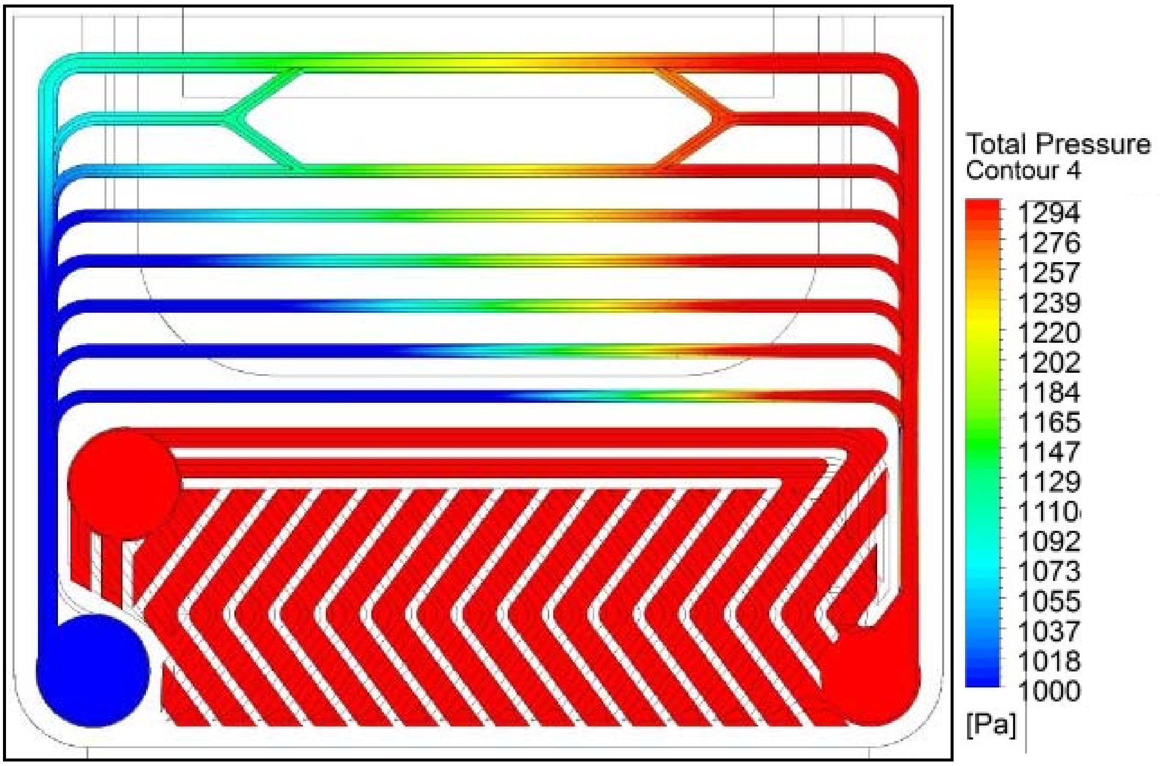

Figure 44 – Pressure contour on water side of the external front module

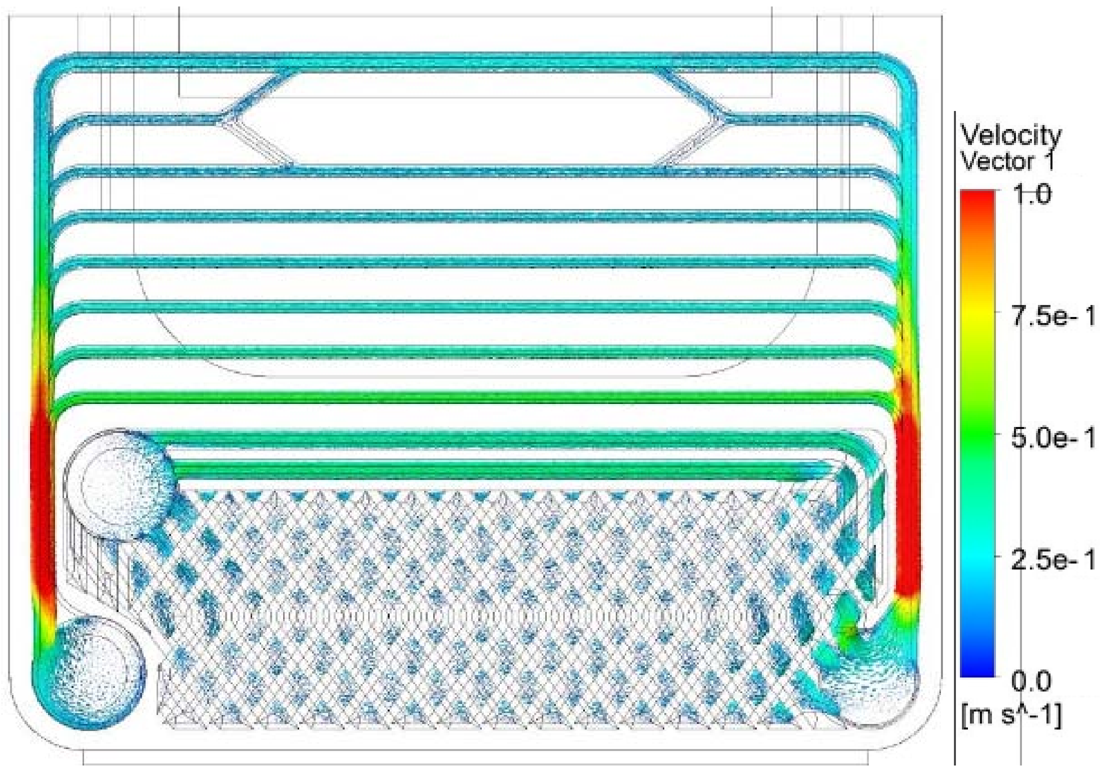

Figure 45 – Velocity contour of water domain in the external front module symmetry plane

Analyzing the above presented results, it is evident that:

- Temperature in the sealing surface is higher than 200 ºC, and therefore lifetime of the seal is compromised;
- Pressure drop of the front module is not balanced with internal modules, as it represents ca. 33% of these;
- Water temperature exceeds 105 ºC in certain locations, which represent high boiling risk;
- The above mentioned water temperature is mainly due to low velocity or stagnant regions around the igniter seal area.

To overcome the above concerns, a new design is proposed with the following design features:

- Change the water path from parallel to a combination of parallel and serial water flow path to increase fluid velocity and therefore turbulence and heat transfer, while reducing hot spots and boiling potential (see Figure 46);
- Add a half channel in the internal plate of the front module for increased cooling of the inner surface of the igniter seal (see Figure 47).

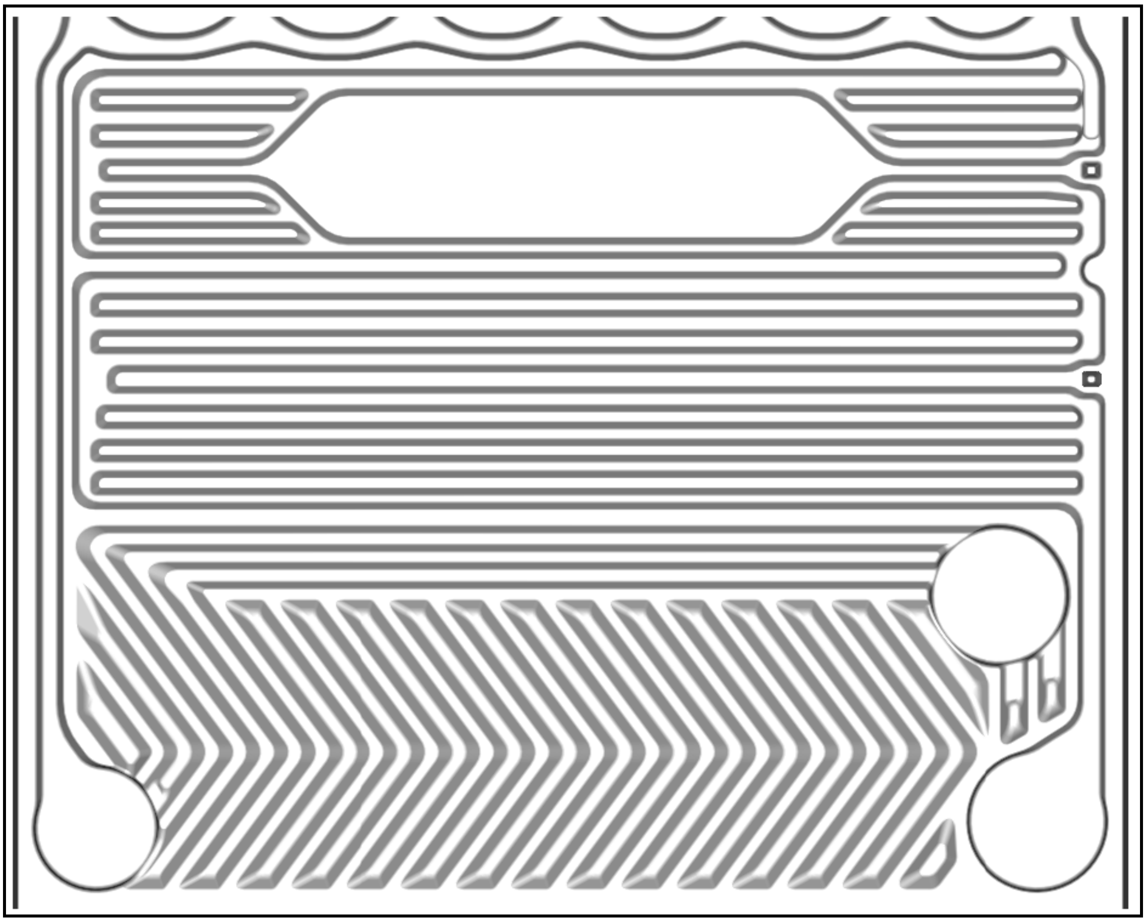

Figure 46 –External plate of front module with combination of parallel and serial water channels

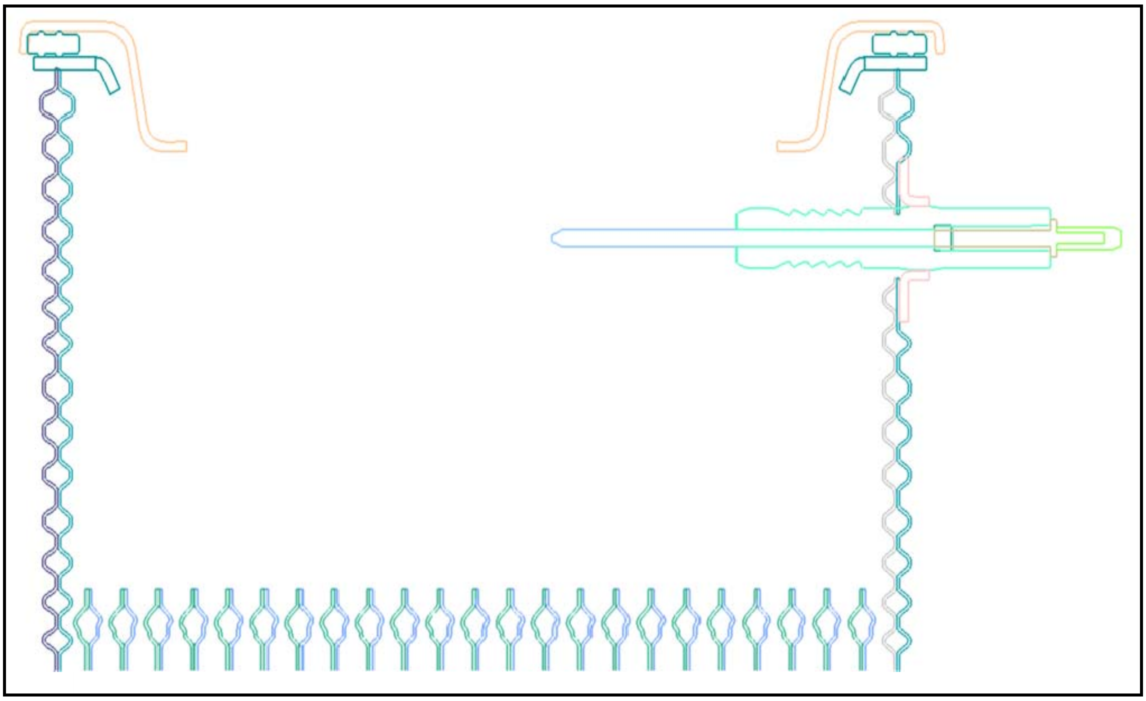

Figure 47 – Position of ignition electrodes in heat cell with half channels added for increased cooling

Several water paths configurations were evaluated in the optimization process of:

- Balancing the pressure drop of front module with internal modules;
- Reducing stagnant or low velocity regions;
- Reduce water temperature below boiling limit;
- Reduce temperature in the seal internal and external surface.

The results presented below represent the best scenario achieved. In summary:

- Material hot spots are reduced to very limited areas of the seal interface with plate. The seal can in theory be designed to avoid touching the hot spots;
- Water temperature is reduced below the boiling limit;
- Internal seal surface temperature is below 80 ºC representing no risk for its lifetime;
- External seal surface temperature exceeds the target limit of 150 ºC, requiring experimental validation before taking additional design measures.

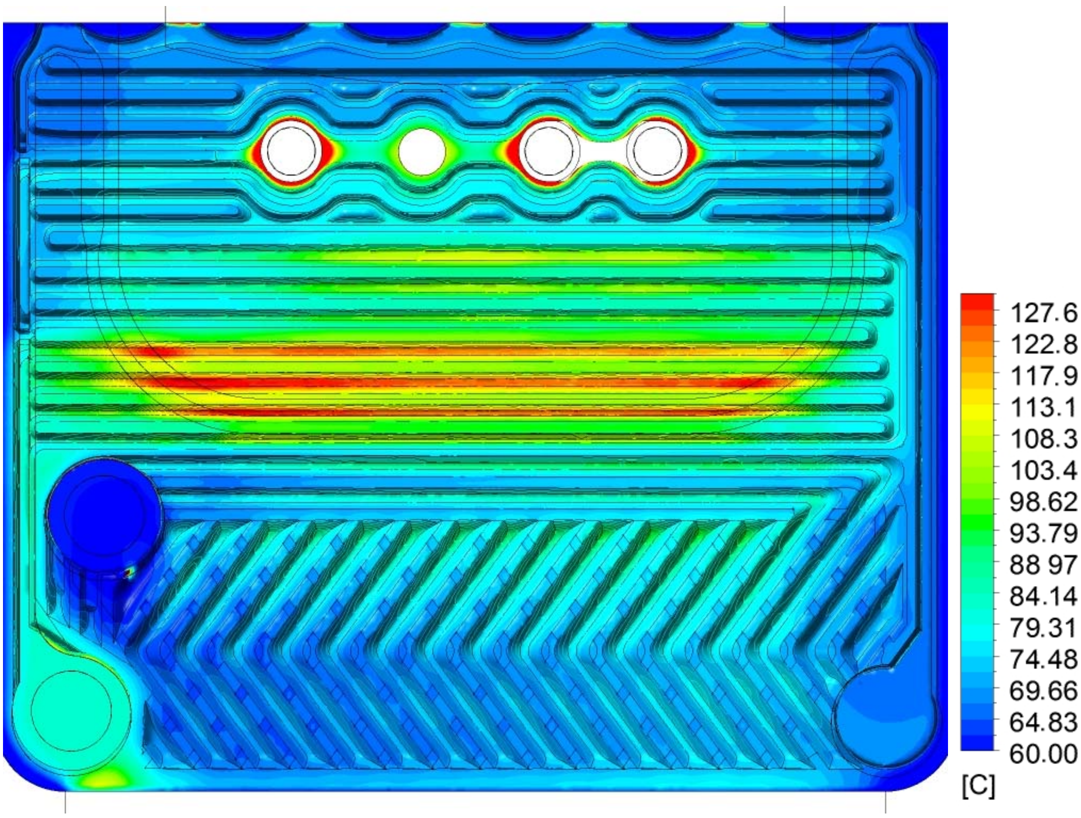

Figure 48 – External plate surface temperature contours

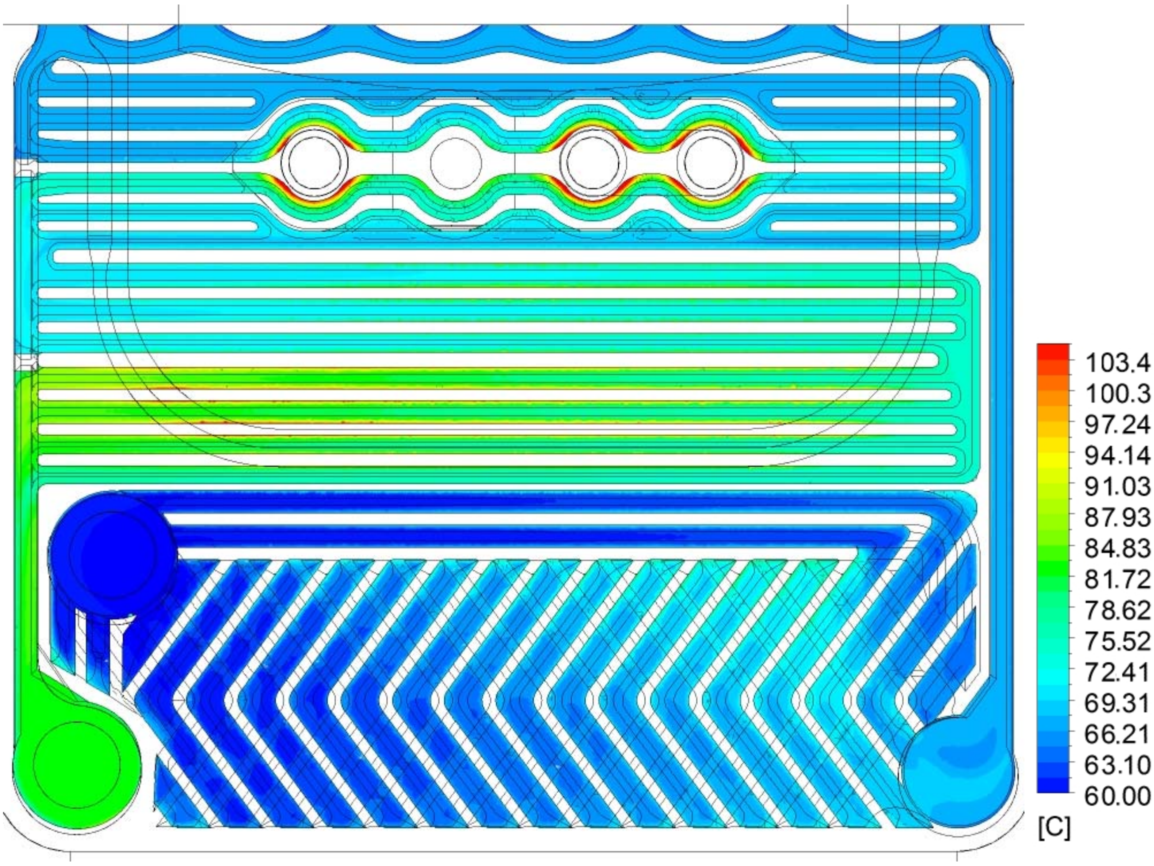

Figure 49 – Water temperature contours in the symmetry plane

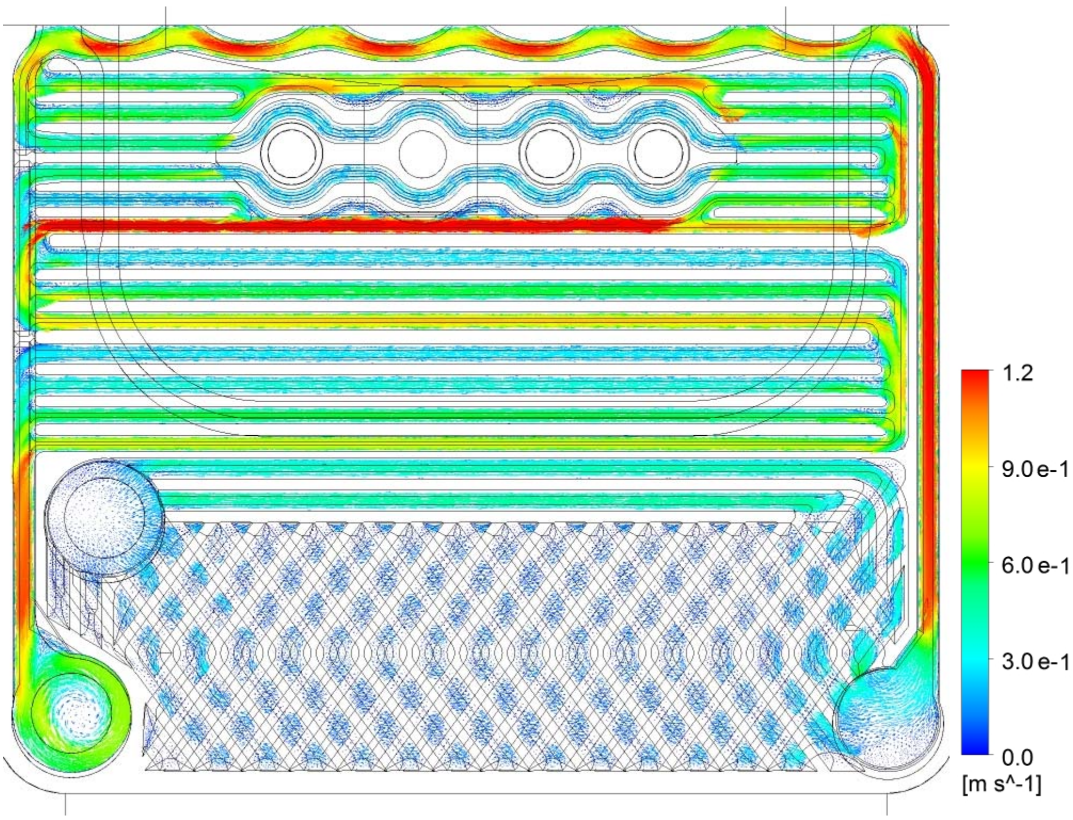

Figure 50 – Water velocity vectors in the symmetry plane

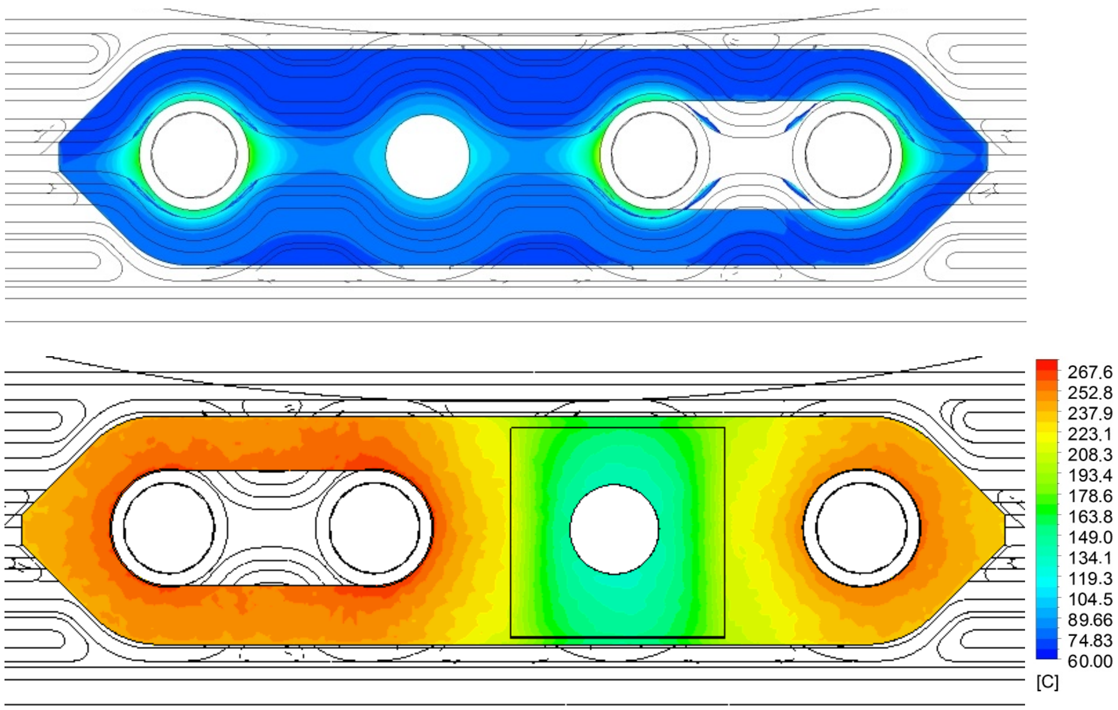

Figure 51 – Seal surface temperatures: internal surface (top) and external surface (bottom)

7 Experimental Set-Up

For the experimental set-up, two different sets of samples were produced:

Sample 1:

- Middle plates according to model nr. 4 presented in Chapter 6;
- Front plates without flow optimization (Figures 42 to 45).

Sample 2:

- Middle plates according to model nr. 4 presented in Chapter 6;
- Front plates with flow optimization to reduce temperatures (Figures 46 to 51).

Plates were produced by stamping, using a prototype tool. The handling and positioning of the raw materials, stainless steel and copper foil, was done manually in the press. The assembly of the modules before brazing in the vacuum oven was also done manually. The brazing process consisted of 3 stages as described below in Figure 52:

- Step 1: brazing of plates together with CH water inlet connection bush and igniter connection bushes (brazing in vacuum oven);
- Step 2: addition of burner and sump plates (brazing in vacuum oven);
- Step 3: addition of CH water outlet connection bush (manual brazing).

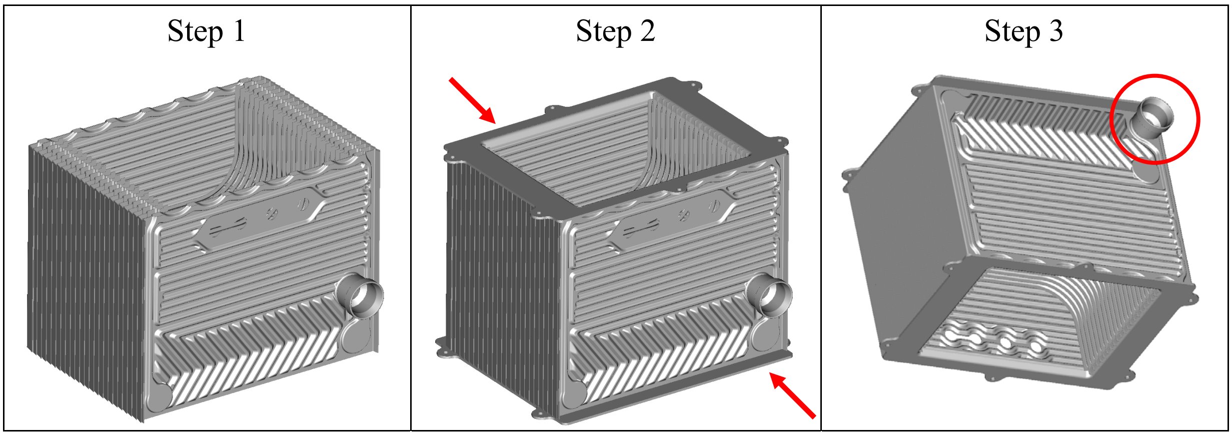

Figure 52 – Steps of sample production

To guarantee the positioning and brazing between the plates, weights have to be used in the oven. However, these must not prevent the flow of copper during the process.

The oven temperature cycle and the amount of weights on top of the samples were not optimized for the production of these samples.

Figure 53 – Positioning of heat exchanger in vacuum oven before the first brazing operation

Figure 54 – Sample 1 used for laboratory validation of concept

Both samples were assembled with standard components used in other combination boiler with the same power range.

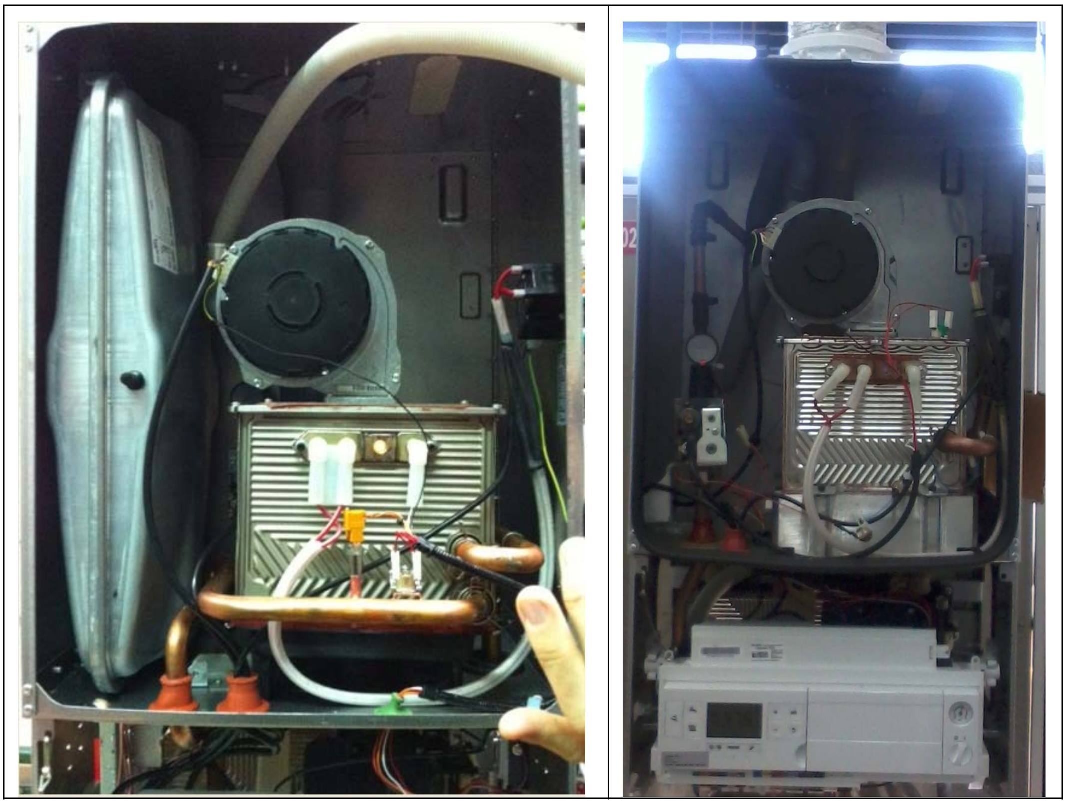

Figure 55 – Complete Heat Cells assembled: Sample 1 (left) and Sample 2 (right)

For the experimental evaluation, the equipment listed in Table 1 was used.

Table 1 – List of laboratory equipment used for laboratory validation of the new concept

Instrument	Manufacturer	Measuring Range	Resolution	Accuracy
CO / CO_2 Analyser	Siemens	CO_2: 0 - 5 / 25 % CO: 0 - 3000 ppm	0,1 % - CO2 0,1 ppm - CO	± 2% rdg
NO_X Analyser	Siemens	NO_X: 0 - 250 ppm	0,1 ppm - NO 0,1 ppm NO2	± 2% rdg
Gas Meter	Instromet	0,04 m^3/h to 25 m^3/h		± 0,5% rdg
Pressure Meter	Yokogawa	0 to 30 bar	0.01 % of span	0,15% of Span
Pressure Sensor	Huba Controls	0 to 25 bar	0.01 % of span	
Pressure Sensor	D P Measurement LTD	0 - 7,00 kPa 0 - 70 mbar 0 - 700 mm H2O	Display / RS323	± (1% rdg + 1 digit)
Temperature Meter	FLUKE	J-Type -200 - 760 °C K-Type -200 - 1370°C	High 0.1 °C of 0.2 °F High 0.1 °C of 0.2 °F	K-type: ± (0.1% rdg + 0.7°C) J-type: ±(0.1% rdg + 0.8°C)
Temperature Meter	Fluke	-210°C to +1300 °C	0,1°C <1000°C 1,0 > 1000 °C	J-K-T-E-N = ± 0.05% rdg + 0.3°C
Temperature Meter	IMPAC electronics GmbH	850 to 1800 °C	4 .. 20 mA	± 1 % Full Scale
Infrared Camera Scanner	FLUKE	-20 °C to + 250 °C		± 2 °C or 2 % (whichever is greater)
Resistance Thermometer	Tempcontrol B.V.	0-120 °C	0,01 °C	± 0,11 °C
Thermocouple Type K	FLUKE	-40 .. 260 °C	-	± 1,1 °C
Thermocouple Type K	FLUKE	-40 .. 260 °C	-	± 1,1 °C
Thermocouple Type K	FLUKE	-40 .. 260 °C	-	± 1,1 °C
Thermocouple Type K	FLUKE	-40 .. 260 °C	-	± 1,1 °C
Thermocouple Type K	FLUKE	-40 .. 260 °C	-	± 1,1 °C
Thermocouple Type K	FLUKE	-40 .. 260 °C	-	± 1,1 °C
Thermocouple Type K	FLUKE	-40 .. 260 °C	-	± 1,1 °C
Thermocouple Type K	FLUKE	-40 .. 260 °C	-	± 1,1 °C
Thermocouple Type K	FLUKE	-40 .. 260 °C	-	± 1,1 °C
Thermocouple Type K	FLUKE	-40 .. 260 °C	-	± 1,1 °C
Thermocouple Type K	FLUKE	-40 .. 260 °C	-	± 1,1 °C
Thermocouple Type K	FLUKE	-40 .. 260 °C	-	± 1,1 °C
Thermocouple Type K	FLUKE	-40 .. 260 °C	-	± 1,1 °C
Thermocouple Type K	FLUKE	-40 .. 260 °C	-	± 1,1 °C
Thermocouple Type K	FLUKE	-40 .. 260 °C	-	± 1,1 °C
Thermocouple Type K	FLUKE	-40 .. 260 °C	-	± 1,1 °C
Thermocouple Type K	FLUKE	-40 .. 260 °C	-	± 1,1 °C
Multimeter	Keithley	10mA 100mA 1A 3A	10nA 100nA 1µA 10µA	± (0,05% rdg + 0,0008 mA) ± (0,05% rdg + 0,08 mA) ± (0,08% rdg + 0,08 mA) ± (0,12% rdg + 0,12 mA)
Multimeter	Fluke	100mV…1000 V 10Ohm - 1 G Ohm	6-1/2 digit	± 0,0044 % measurement 0,015 Ohm… 1 Ohm
Multimeter	Fluke	100mV…1000 V 10Ohm - 1 G Ohm	6-1/2 digit	± 0,0044 % measurement 0,015 Ohm… 1 Ohm
Oscilloscope	Fluke	200 MHz 2.5 GS/s	8 bits / input	± 1,5 %
Current Clamp	ABB	200 A AC	1 A .. = 10 mV AC	1 % full scale at 200 A 5 % full scale at 100 mA
Scale	Mettler Toledo	0 - 5000 gr.	0,1	
pH Meter	WTW	0 - 14	0,01	± 0,01

8 Results and Discussion

8.1 Sample 1

The tests performed with Sample 1, according description in previous chapter, are the following:

- Nominal Heat Input according to EN 483:1999 +A4:2007 (Tests through modulation range including flue gas emissions and temperature characterization);
- Thermal Efficiency according to EN 483:1999 +A4:2007 (Tests through modulation range);
- Assessment of Hot Water Deliveries Performance according to EN13203-1:2006;
- Assessment of Energy Consumption according to cycle nr. 2 from EN13203-2:2006;
- Assessment of critical temperatures in heat cell.

Unless otherwise stated, all tests were performed with appliance in steady state, using reference gas G20 and with 0.5 m of concentric flue gas pipe ∅ 60-100 mm installed (flue gas in inner pipe with 60 mm diameter and fresh air in outer pipe with 100 mm diameter).

8.1.1 Heat Input, Efficiencies and Flue Gas Emissions through the Modulation Range

For the determination of the heat inputs and efficiencies, the boiler is supplied with the reference gas (G20 –methane) at the normal gauge or relative pressure for this test type (20 mbar).

According to EN 483, test may begin once the boiler is at thermal equilibrium and the return and flow water temperatures are constant. Following good laboratory practices, the stabilization time used in all tests of present report is 15 minutes. Readings of the water flow rate return and flow water temperatures are taken periodically so as to obtain a sufficiently accurate average (1 second chosen for all tests in present report) and the test runs for 10 minutes.

The volumetric gas rate obtained under these conditions (p_a, p_g, t_g, d) shall be corrected as if the test had been carried out under the reference test conditions (1013.25 mbar, 15 ºC, dry gas) and the corrected heat input is calculated using the following formula [9]:

$$Q_c = H_i \times \frac{10^3}{3600} \times V \times \sqrt{\frac{1013.25 + p_g}{1013.25} \times \frac{p_a + p_g}{1013.25} \times \frac{288.15}{273.15 + t_g} \times \frac{d}{d_r}} \qquad (1)$$

where,

Q_C is the corrected heat input (1013.25 mbar and 15 ºC, dry gas) in kilowatt (kW);

V is the measured volumetric gas rate in cubic meters per hour (m^3/h);

Hi is the net calorific value of dry reference gas at 15 ºC, 1013.25 mbar in mega Joule per cubic meter (MJ/m^3);

t_g is the gas temperature at the meter in degrees Celsius (ºC);

d is the relative density of the test gas;

d_r is the relative density of the reference gas;

p_g is the relative gas pressure at the meter in milibar (mbar);

p_a is the ambient pressure at the time of the test in milibar (mbar).

The useful efficiency is determined by means of the following formula [9]:

$$\eta_u = \frac{4.186 \times m \times \left(T_2 - T_1\right)}{10^3 \times V_{r(10)} \times H_i} \times 100 \tag{2}$$

where,

η_u is the useful efficiency in percentage;

m is the corrected quantity of water expressed in kilogram (kg);

$V_{r(10)}$ is the gas consumption in m^3 measured during the test corrected to 15 ºC, 1013.25 mbar;

T_2 is the flow temperature from the heat cell in degrees Celsius (ºC);

T_1 is the return temperature to the heat cell in degrees Celsius (ºC).

The measurement uncertainties are chosen in a way which ensures a total uncertainty in the efficiency measurement of ± 2%. The flue gas emissions are registered in all tests and corrected as follows [9]:

$$CO_{DAF} = (CO)_{Measured} \times \frac{(CO_2)_{Max}}{(CO_2)_{Measured}} \tag{3}$$

$$NO_{X\,DAF} = (NO_X)_{Measured} \times \frac{(CO_2)_{Max}}{(CO_2)_{Measured}} \tag{4}$$

where,

CO_{DAF} is the carbon monoxide of the dry air free combustion products in percentage;

$NO_{X\,DAF}$ is the nitrogen oxides of the dry air free combustion products in percentage;

$(CO)_{Measured}$ is the measured concentration of carbon monoxide in percentage;

$(NO_X)_{Measured}$ is the measured concentration of nitrogen oxides in percentage;

$(CO_2)_{Max}$ is the maximum carbon dioxide concentration of the dry air free combustion products in percentage;

$(CO_2)_{Measured}$ is the measured concentration of carbon dioxide in percentage.

The maximum CO_2 percentage in the flue gases for G20 is 11.7% (v/v).

The sample was evaluated in 2 different flow/return conditions: 80/60 and 50/30. These are the standard efficiencies declared in end user manuals and they enable the evaluation of the thermal efficiency with and without condensation. In 80/60 conditions, the heat cell feeds the heating system with water at 80 ºC (flow temperature) and receives cooled water at 60 ºC (return temperature). The same principle applies to 50/30. The results are presented in the Tables 2 and 3 and in the Figures 56 to 59.

Table 2 – Results of sample 1 under 80/60 test conditions

Fan Speed [rpm]	2880	3450	4320	5160	6600	7100
Ambient Temperature [ºC]	23.2 ºC	23.8 ºC	23.1 ºC	22.9 ºC	22.4 ºC	22.5 ºC
Ambient Pressure [mbar]	996.7 mbar	1000.2 mbar	999.9 mbar	999.8 mbar	998.8 mbar	999.4 mbar
Gas Temperature [ºC]	22.8 ºC	23.9 ºC	25.1 ºC	25.6 ºC	25.3 ºC	26.0 ºC
CH Mode	80/60	80/60	80/60	80/60	80/60	80/60
Average Flow Temperature [ºC]	79.5 ºC	79.9 ºC	80.0 ºC	80.1 ºC	80.0 ºC	80.2 ºC
Average Return Temperature [ºC]	59.6 ºC	59.8 ºC	60.0 ºC	60.0 ºC	60.0 ºC	60.1 ºC
Average ΔT [ºC]	19.9 ºC	20.2 ºC	20.0 ºC	20.0 ºC	19.9 ºC	20.1 ºC
CH Water Rate [kg/h]	378 kg/h	468 kg/h	600 kg/h	729 kg/h	913 kg/h	942 kg/h
$Q_{C\ Nominal}$ [kW]	24.0 kW	24.0 kW	24.0 kW	24.0 kW	24.0 kW	24.0 kW
$Q_{C\ Measured}$ [kW]	**9.4 kW**	**11.7 kW**	**14.9 kW**	**18.1 kW**	**22.6 kW**	**24.0 kW**
$Q_{C\ Meas}$ / $Q_{C\ Nom}$ [%]	39%	49%	62%	75%	94%	100%
Q_i [kW]	9.2 kW	11.4 kW	14.5 kW	17.7 kW	22.1 kW	23.5 kW
Efficiency [%]	**96.6 %**	**97.0 %**	**97.5 %**	**97.4 %**	**97.1 %**	**97.1 %**
Pm [kW]	8.8 kW	11.1 kW	14.2 kW	17.2 kW	21.4 kW	22.8 kW
CO_2 [%]	**8.5 %**	**8.9 %**	**9.1 %**	**9.2 %**	**9.3 %**	**9.3 %**
CO [ppm]	14 ppm	23 ppm	35 ppm	45 ppm	56 ppm	61 ppm
$CO_{(DAF)}$ [ppm]	**19 ppm**	**30 ppm**	**45 ppm**	**57 ppm**	**71 ppm**	**77 ppm**
NO_X [ppm]	5.7 ppm	8.6 ppm	9.9 ppm	10.5 ppm	10.0 ppm	10.5 ppm
$NO_{X\ (DAF)}$ [ppm]	**8 ppm**	**11 ppm**	**13 ppm**	**13 ppm**	**13 ppm**	**13 ppm**
Flue Gas Temperature [ºC]	**57.1 ºC**	**60.3 ºC**	**61.7 ºC**	**67.0 ºC**	**73.3 ºC**	**75.0 ºC**
Dew Point Temperature [ºC]	53.4 ºC	54.2 ºC	54.6 ºC	54.8 ºC	55.0 ºC	55.0 ºC

Table 3 – Results of sample 1 under 50/30 test conditions

Fan Speed [rpm]	2220	2880	3450	4320	5160	6600	7100
Ambient Temperature [ºC]	22.5 ºC	22.4 ºC	22.9 ºC	22.9 ºC	22.9 ºC	23.0 ºC	23.0 ºC
Ambient Pressure [mbar]	992.7 mbar	992.9 mbar	992.9 mbar	992.7 mbar	991.1 mbar	991.0 mbar	991.1 mbar
Gas Temperature [ºC]	22.6 ºC	23.1 ºC	23.5 ºC	23.8 ºC	23.5 ºC	24.0 ºC	24.0 ºC
CH Mode	50/30	50/30	50/30	50/30	50/30	50/30	50/30
Average Flow Temperature [ºC]	49.8 ºC	50.0 ºC	49.9 ºC	49.9 ºC	50.0 ºC	49.8 ºC	50.0 ºC
Average Return Temperature [ºC]	30.0 ºC	29.9 ºC	30.0 ºC	30.0 ºC	29.9 ºC	29.9 ºC	30.1 ºC
Average ΔT [ºC]	19.8 ºC	20.0 ºC	19.9 ºC	20.0 ºC	20.1 ºC	20.0 ºC	19.9 ºC
CH Water Rate [kg/h]	313 kg/h	417 kg/h	511 kg/h	646 kg/h	789 kg/h	1004 kg/h	1085 kg/h
$Q_{C\ Nominal}$ [kW]	24.0 kW	24.0 kW	24.0 kW	24.0 kW	24.0 kW	24.0 kW	24.0 kW
$Q_{C\ Measured}$ [kW]	**6.9 kW**	**9.4 kW**	**11.4 kW**	**14.4 kW**	**17.8 kW**	**22.8 kW**	**24.3 kW**
$Q_{C\ Meas}$ / $Q_{C\ Nom}$ [%]	29%	39%	47%	60%	74%	95%	101%
Q_i [kW]	6.7 kW	9.1 kW	11.1 kW	14.1 kW	17.4 kW	22.2 kW	23.7 kW
Efficiency [%]	**106.9 %**	**106.8 %**	**106.7 %**	**106.5 %**	**105.9 %**	**105.2 %**	**105.0 %**
Pm [kW]	7.2 kW	9.8 kW	11.9 kW	15.0 kW	18.4 kW	23.3 kW	24.9 kW
CO_2 [%]	**8.45 %**	**8.70 %**	**8.84 %**	**8.92 %**	**9.05 %**	**9.12 %**	**9.15 %**
CO [ppm]	11.3	16.7	22.6	32.0	47.0	68.5	75.0
CO $_{(DAF)}$ [ppm]	**16 ppm**	**22 ppm**	**30 ppm**	**42 ppm**	**61 ppm**	**88 ppm**	**96 ppm**
NO_X [ppm]	5.6 ppm	7.3 ppm	8.2 ppm	8.6 ppm	10.0 ppm	10.8 ppm	11.2 ppm
$NO_{X\ (DAF)}$ [ppm]	**8 ppm**	**10 ppm**	**11 ppm**	**11 ppm**	**13 ppm**	**14 ppm**	**14 ppm**
Flue Gas Temperature [ºC]	**33.1 ºC**	**33.8 ºC**	**35.0 ºC**	**37.1 ºC**	**40.0 ºC**	**44.5 ºC**	**46.0 ºC**
Dew Point Temperature [ºC]	33.1 ºC	33.8 ºC	35.0 ºC	37.1 ºC	40.0 ºC	44.5 ºC	44.5 ºC

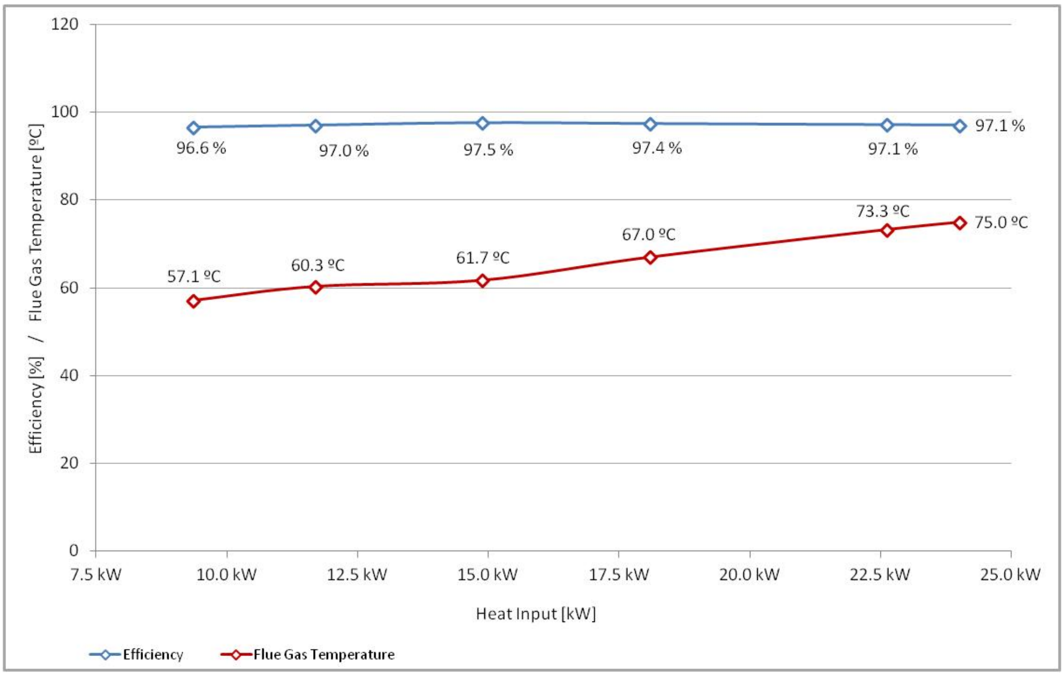

Figure 56– Efficiency and flue gas temperature through modulation range measured in 80/60 conditions

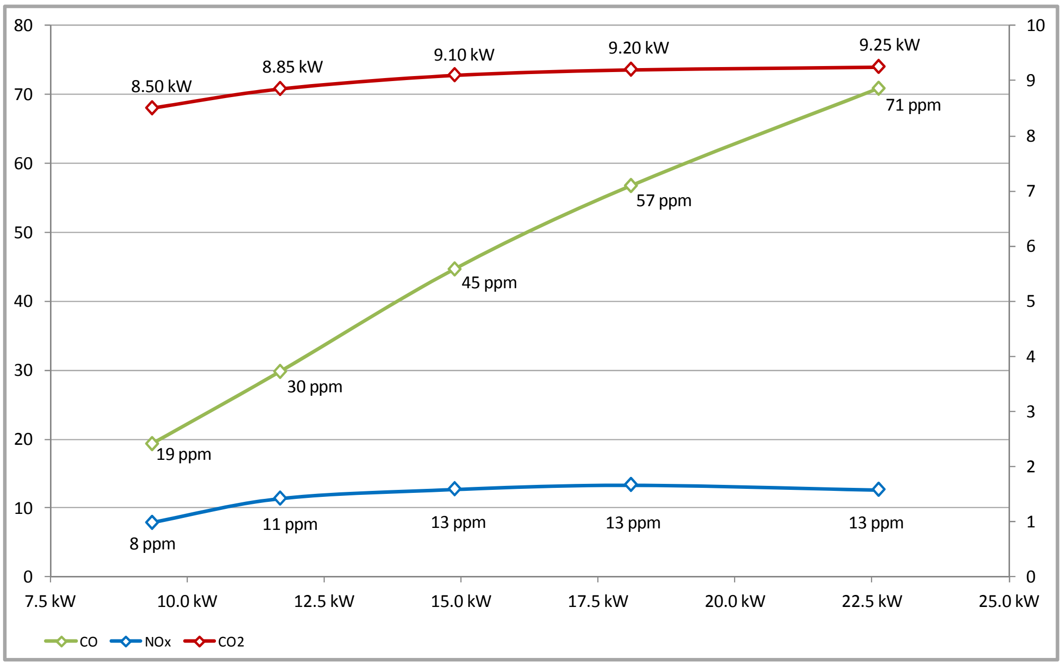

Figure 57– Corrected flue gas emissions through modulation range measured in 80/60 conditions

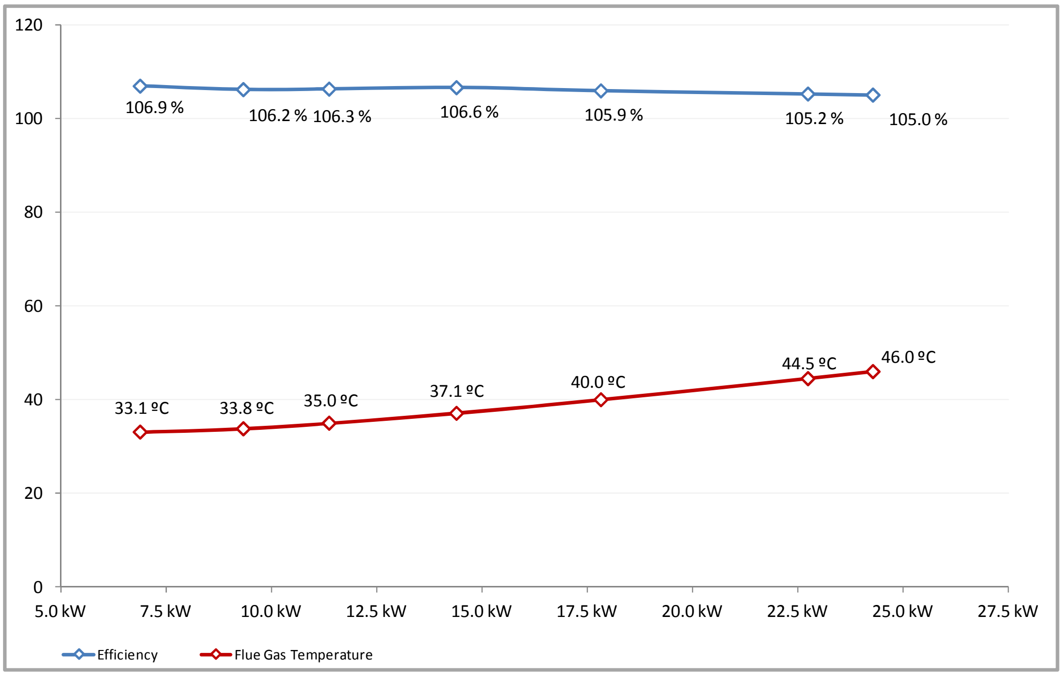

Figure 58– Efficiency and flue gas temperature through modulation range measured in 50/30 conditions

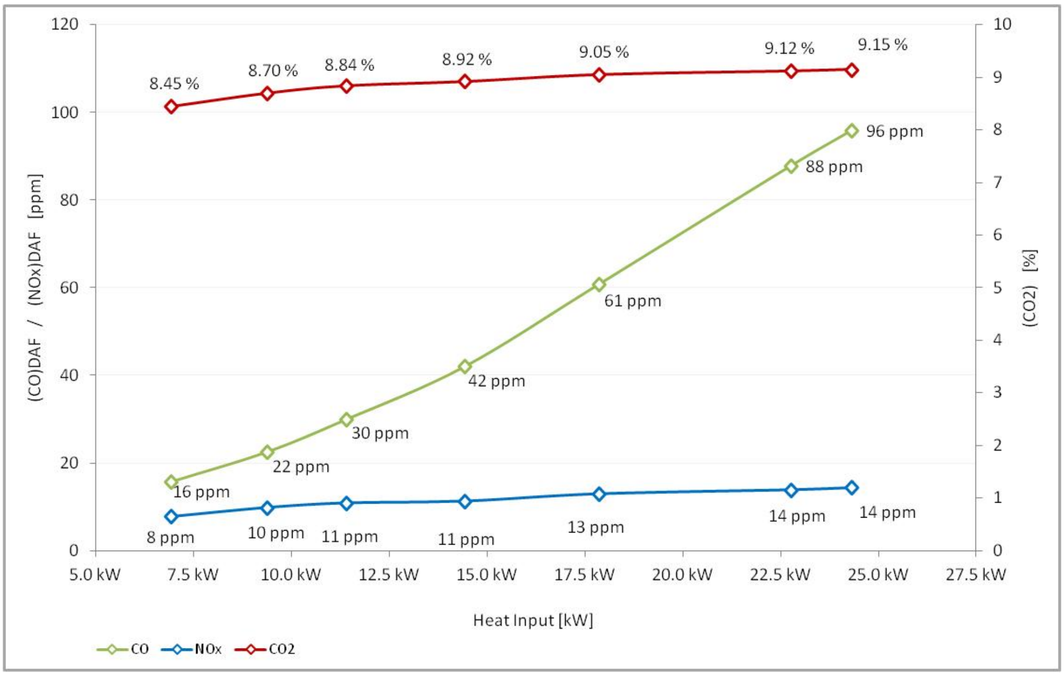

Figure 59– Corrected flue gas emissions through modulation range measured in 50/30 conditions

8.1.2 – Assessment of Hot Water Deliveries Performance (according to EN 13203-1:2006)

The performance in delivery of domestic hot water for a selected variety of uses is described in EN 13203-1 which enables it to be assessed both in qualitative and quantitative terms. It also gives a system for presenting the information to the user.

The domestic hot water function of domestic hot water appliances is characterized in two different ways:

- Firstly, according to the domestic hot water specific rates, the tapping capability and the corresponding uses;
- Secondly, according to the quality of the domestic hot water produced, obtaining a number of stars corresponding to a determined level of performance.

For the determination of the boiler Specific Rate *D*, the test follows the sequence:

- Water delivery of 10 min duration;
- 20 min period of no water delivery;
- Further water delivery of 10 min duration.

Measurements of temperature and flow rate are made and recorded, at intervals not exceeding 2 s. A plot of temperature against time is made, to obtain the mean water temperature rise during each delivery.

For each delivery the corresponding Specific Rate is calculated by [10]:

$$D_i = \frac{m_{i(10)}}{10} \times \frac{\Delta T}{30} \tag{5}$$

where,

D_i is the domestic water rate corresponding to a mean temperature rise of 30 K in liters per minute (l/min) that the boiler can supply in two successive delivery periods;

$m_{i(10)}$ is the quantity of water collected with a minimum temperature rise of 30 K in liters (l);

ΔT is the mean temperature rise of the collected water, in Kelvins (K).

If the difference between D_1 and D_2 does not exceed numerically 10 % of their average value then [10]:

$$D = \frac{D_1 + D_2}{2} \tag{6}$$

The Kitchen Specific Rate is then calculated by the formula [10]:

$$D_c = D \times \frac{30}{45} \tag{7}$$

where,

D_C is the kitchen specific rate, i.e. the domestic water rate corresponding to a mean temperature rise of 45 K that the boiler can supply.

The values obtained for these are: D = 12.7 l/min and D_C = 8.5 l/min. These values, Specific Rate and Kitchen Specific Rate, will be used as inputs for the remaining following tests [10]:

- Waiting time;
  (time taken to reach 90% of the domestic hot water temperature rise of 45 K)
- Temperature variation according to water flow rate;
  (variation of the hot water temperature consequent upon variations of the water flow rate)
- Temperature variation at constant water flow rate;
  (difference between the minimum and maximum water temperatures that can occur during delivery at a constant water rate with a constant inlet temperature)
- Temperature stabilization time;
  (time taken to obtain a predetermined fluctuation, following a rapid variation of the water flow rate)
- Minimum nominal water rate;
  (lowest water rate stated by the manufacturer maintaining a stable temperature)
- Temperature fluctuation between successive deliveries.
  (maximum domestic hot water temperature difference between successive deliveries)

Depending on the performance on each one of the tests above, a weighted performance factor f_i is calculated according to the operating conditions explained in Table 4.

Table 4 – Symbols of particular performance and weighting criteria [10] (clause 3 refers to EN 13203-1:2006 – Chapter 3 – Terms and Definitions)

Particular performance criterion	Symbol (as in clause 3)	Particular performance factor (f_i)				Weighting coefficient a_i
		0	1	2	3	
Waiting time	t_m	> 60 s	≤ 60 s	≤ 35 s	≤ 5 s	4
Temperature variation according to water rate	ΔT_1	> 10 K	≤ 10 K	≤ 5 K	≤ 2 K	3
Temperature fluctuation at constant water rate	ΔT_2	> 5 K	≤ 5 K	≤ 3 K	≤ 2 K	3
Temperature stabilisation time	t_s	≥ 60 s	< 60 s	< 30 s	< 10 s	2
Minimum nominal water rate	D_m	> 6 l/min	≤ 6 l/min	≤ 4 l/min	≤ 2 l/min	1
Temperature fluctuation between successive deliveries	ΔT_3	> 20 K	≤ 20 K	≤ 10 K	≤ 5 K	1

The overall performance factor is then calculated as follows [10]:

$$F = \sum_{i=1}^{n} a_i \times f_i \tag{8}$$

Depending on the value obtained, the overall performance factor *F* is used to classify the performance of the domestic hot water delivered, as indicated in Table 5 presented below:

Table 5 – Classification according to the overall performance factor *F* [10]

Label	Value of the factor *F*
_ _ _	< 14 points
* _ _	14 to 27 points
* * _	28 to 39 points
* * *	≥ 40 points with particular factors ≥ 2

The results achieved with the sample heat cell under analysis are presented below. The legend for figures 60 to 66 follows the explanation below:

- Orange: Domestic water outlet temperature (thermocouples) in degrees Celsius (ºC)
- Red: Domestic water outlet temperature (PT100) in degrees Celsius (ºC)
- Dark Blue: Domestic water flow rate in liters per minute (l/min)
- Light Blue: Domestic water inlet temperature (PT100) in degrees Celsius (ºC)

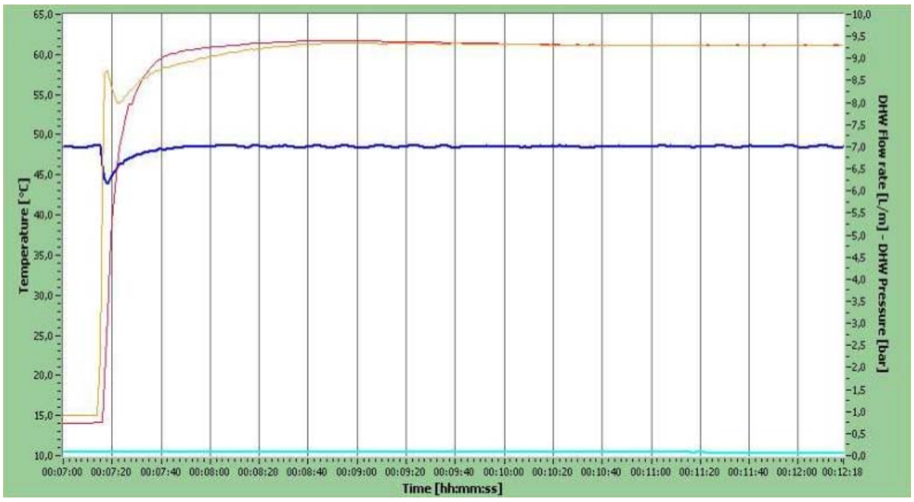

Figure 60– Waiting time test sequence t_1 = 2.0 s

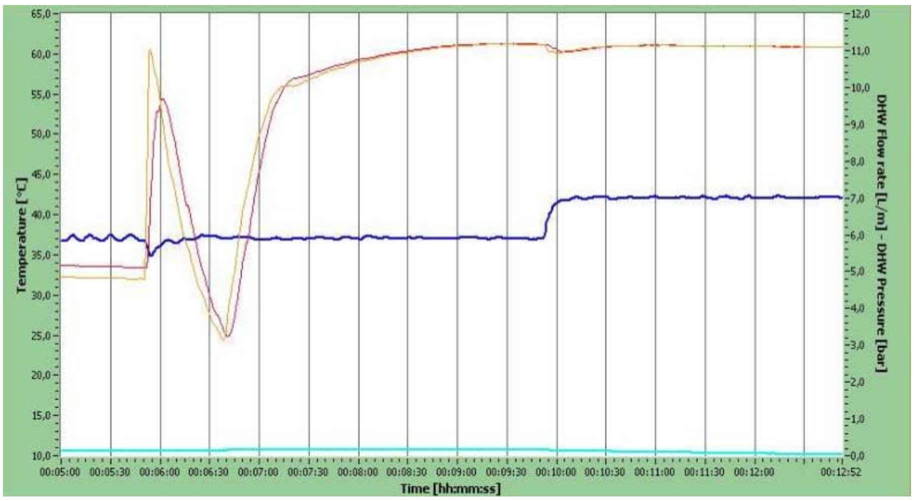

Figure 61 – Temperature variation according to the water flow rate test sequence, T_{1m} = 60.4 °C T_{2m} = 61.0 °C ΔT_m = 0.5 °C

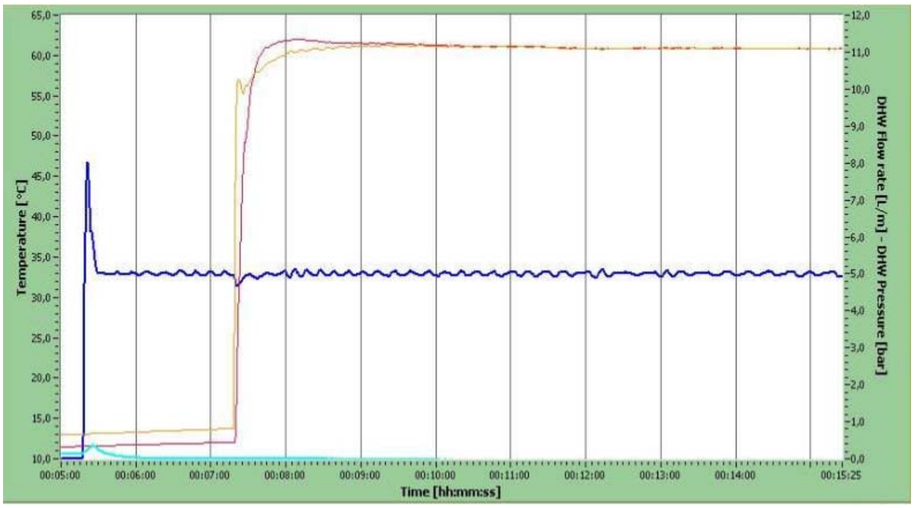

Figure 62 – Temperature fluctuation at constant water flow rate test sequence with water flow of 5 l/min and ΔT_{2max} = 0.6 °C

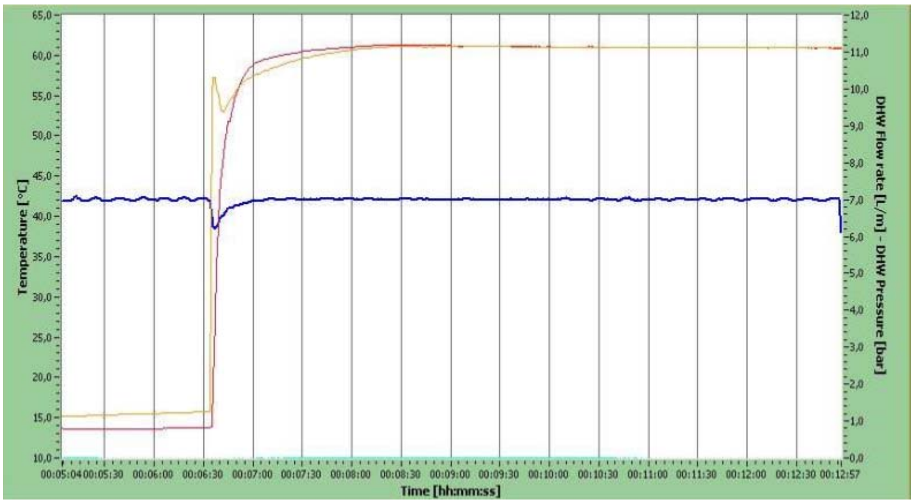

Figure 63 – Temperature fluctuation at constant water flow rate test sequence with water flow of 5 l/min and ΔT_{1max} = 0.2 °C

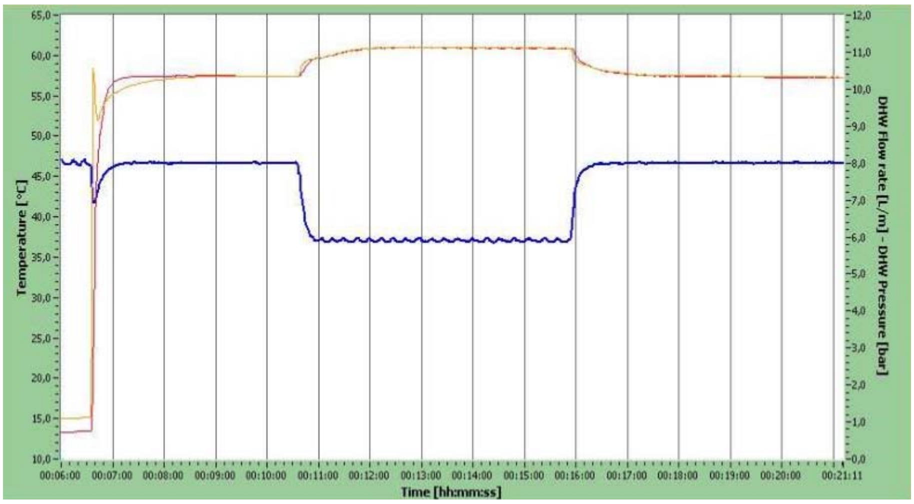

Figure 64 – Temperature stabilization test sequence, t_{S1} = 0 s t_{S2} = 0 s.

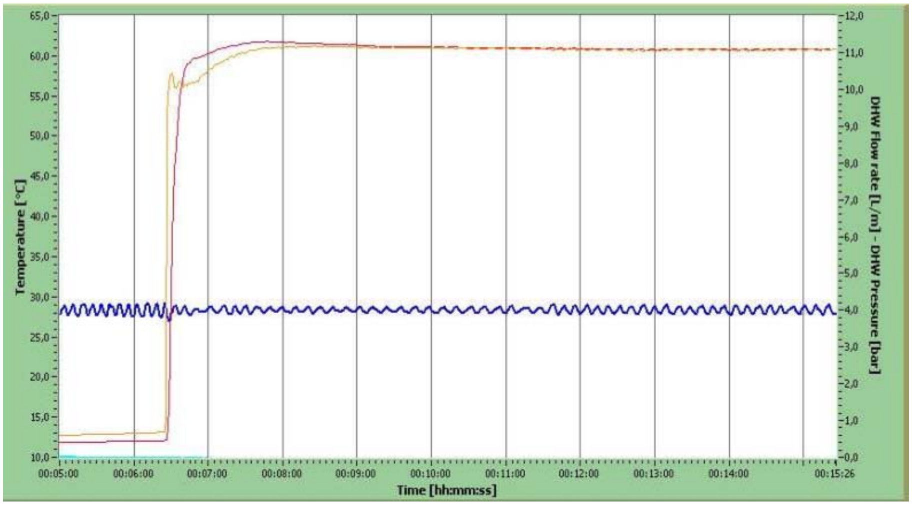

Figure 65 – Minimum nominal water rate test sequence for 4l/min and maximum temperature fluctuation of +0.4 – (–0.3) = 0.7 °C

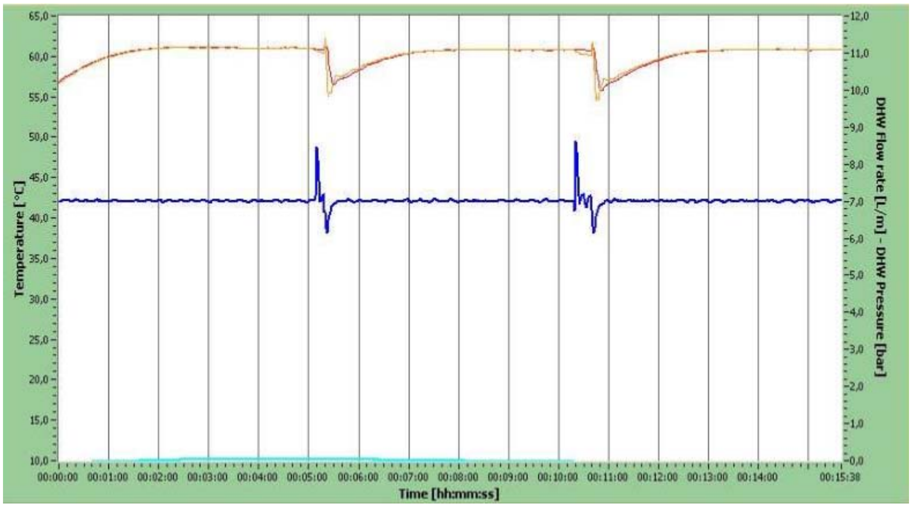

Figure 66 – Temperature fluctuation between successive deliveries ΔT_3 = 0.7 °C

The summary of all tests above is presented in the Table 6 below. The overall performance factor F is 41 points (out of 42 points) and therefore the rating is 3 Stars.

Table 6 – Summary of Comfort Tests Results

Performance Criterion	Sample 1	
	Measured	Score
T_m	2.0 s	3 × 4 = 12
ΔT_1	0.5 ºC	3 × 3 = 9
ΔT_2	0.6 ºC	3 × 3 = 9
t_S	0.0 s	3 × 2 = 6
D_m	4.0 l/min	2 × 1 = 2
ΔT_3	0.7 ºC	3 × 1 = 3
Σ	---	41

8.1.3 – Assessment of Daily Energy Consumption (according to EN 13203-2:2006)

The European Standard EN 13203-2 sets out a method for assessing the energy performance of heating appliances (instantaneous and storage appliances; waters-heaters and combination boilers with heat input not exceeding 70 kW, and hot water storage capacity not exceeding 300 l). It defines a number of daily tapping cycles for each domestic hot water use, kitchen, shower, bath and a combination of these, together with corresponding test procedures, enabling the energy performances of different gas-fired appliances to be compared and matched to the needs of the user [11].

All patterns in the standard define a 24 h measurement cycle and within that cycle the starting times and the total energy content (in kWh equivalent of hot water tapped) of each draw-off are defined.

Furthermore, the draw-off can be characterized in two ways, either [11]:

- "Basin" type draw-off (bath, dishwasher) versus "continuous flow" draw-offs (shower, hand wash etc.).

  The aim of the former is to arrive at an average temperature of the tub, so all supplied energy can be considered useful from the very beginning of the draw-off (minimum useful ΔT is 0 K).

  The latter start to be useful only from a certain temperature (minimum useful temperature increase is 15 K lower than the desired temperature), or
- "Kitchen" type draw-off which is carried out with a water temperature increase of 45 K.

The tapping flow rates used to perform the different types of tapping of each of the five tapping cycles defined in the standard are given in Table 7 below. If by design of the appliance the test cannot be carried out with these low flow rates, the minimum flow rate for the ignition of the appliance is taken. If by design the appliance is fitted with an excess flow valve, the tests are carried out with this excess flow rate.

Table 7 – Tapping flow rates used in tapping cycles [11]

Type of tapping	Energy (kWh)	Hot water flow rates corresponding to a temperature rise of 45 K (l/min)
Household cleaning	0.105	3 ± 0,5
Small	0.105	3 ± 0,5
Floor cleaning	0.105	3 ± 0,5
Dish washing	0.315	4 ± 0,5
Dish washing	0.420	4 ± 0,5
Dish washing	0.735	4 ± 0,5
Large (cycle n°1)	0.525	4 ± 0,5
Shower	1.400	6 ± 0,5
Shower (cycles n°4 et n°5)	1.800	6 ± 0,5
Bath	3.605	10 ± 0,5
Bath (cycle n°4)	4.420	10 ± 0,5
Shower + Bath (cycle n°5)	6.240	16 ± 0,5

The tapping cycle chosen to evaluate the proposed heat cell concept, is the tapping cycle nr. 2. This was chosen because it is the most representative in the UK market, currently one of the top performers in the world. The details of the cycle are as shown in Table 8 below:

Table 8 – Details of tapping cycle number 2 [11]

	Start (h.min)	Energy (kWh)	Type of delivery	ΔT desired (K), to be achieved during tapping	Min. ΔT (K), = start of counting useful energy
1	07.00	0.105	Small		15
2	07.15	1.400	Shower		30
3	07.30	0.105	Small		15
4	08.01	0.105	Small		15
5	08.15	0.105	Small		15
6	08.30	0.105	Small		15
7	08.45	0.105	Small		15
8	09.00	0.105	Small		15
9	09.30	0.105	Small		15
10	10.30	0.105	Floor cleaning	30	0
11	11.30	0.105	Small		15
12	11.45	0.105	Small		15
13	12.45	0.315	Dish washing	45	0
14	14.30	0.105	Small		15
15	15.30	0.105	Small		15
16	16.30	0.105	Small		15
17	18.00	0.105	Small		15
18	18.15	0.105	Household cleaning		30
19	18.30	0.105	Household cleaning		30
20	19.00	0.105	Small		15
21	20.30	0.735	Dish washing	45	0
22	21.15	0.105	Small		15
23	21.30	1.400	Shower		30
	Total	5.845			

The useful energy recovered by the water, Q_{H2O} (kWh) is given by the following equation [11]:

$$Q_{H2O} = 1.163 \times 10^{-3} \sum_{i=1}^{n} \int_{0}^{t1} d_i \times \Delta T_i(t)\, dt \qquad (9)$$

where,

n is the number of tappings;

d_i is the water rate delivered in litre per minute (l/min);

$\Delta T_i\,(t)$ is the instantaneous temperature rise during the tapping, in Kelvin (K);

t_i is the tapping duration of the useful water, in minute (min).

The daily energy consumption is calculated according to the following equations [11]:

$$Q_{gas} = \frac{V_g \times K \times H_i \times Q_{TOT}}{Q_{H2O}} \tag{10}$$

where,

Q_{gas} is the daily energy contributed in kilowatthour (kWh) calculated using H_i;

V_g is the gas consumption in cubic meters (m$^{3)}$;

H_i is the net calorific value in kWh/m3 (at 15 °C and 1 013.25 mbar);

Q_{TOT} is the total delivered energy of used tapping cycle in kWh;

and

$$K = \frac{p_a + p_g}{1013.25} \times \frac{288.15}{T_g + 273.15} \tag{11}$$

where,

p_a is the atmospheric pressure, in milibar (mbar);

p_g is the relative gas pressure, in milibar (mbar);

T_g is the gas temperature, in degrees Celsius (°C).

The boiler efficiency (ε) will then be calculated as:

$$\varepsilon = \frac{Q_{H2O}}{Q_{Gas} + 2.5 \times Q_{Electric}} \tag{12}$$

where,

$Q_{Electric}$ is the electrical consumption during the tapping efficiency test in kWh.

The sample was installed with 0.35 m of ∅ 60-100 concentric flue, operated with G20 and adjusted to:

- DHW (Domestic Hot Water) Maximum Heat Input = 36 kW with CO_2 of 9.2 %
- Minimum Heat Input = 7.0 kW with CO_2 of 8.6 %

The results obtained with the sample under evaluation are the following:

Table 9 – Tapping results presented cycle by cycle

$T_{H2O\ In}$ [ºC]	$T_{H2O\ Out}$ [ºC]	ΔT_{H2O} [ºC]	DWFR [l/min]	Q_{Useful} [kWh]	V_{Gas} [m^3]	Q_{Gas} [kWh]	ε [%]
10.43	40.18	29.75	2.94	0.1091	0.0410	0.4	28%
9.94	58.48	48.54	5.35	1.4072	0.1580	1.5	94%
9.49	59.41	49.92	2.97	0.1078	0.0090	0.1	127%
9.39	53.26	43.87	2.86	0.1101	0.0220	0.2	53%
9.63	57.79	48.16	2.85	0.1100	0.0110	0.1	106%
10.71	55.58	44.87	2.88	0.1072	0.0170	0.2	67%
10.04	57.09	47.05	2.90	0.1070	0.0130	0.1	87%
9.90	54.86	44.96	2.86	0.1083	0.0190	0.2	60%
10.80	54.93	44.13	2.91	0.1081	0.0180	0.2	64%
10.20	50.85	40.65	2.92	0.1083	0.0230	0.2	50%
9.03	50.86	41.83	2.93	0.1089	0.0240	0.2	48%
10.33	57.98	47.65	2.87	0.1099	0.0120	0.1	97%
10.54	56.99	46.45	3.58	0.3199	0.0480	0.5	71%
10.62	47.26	36.64	2.94	0.1073	0.0300	0.3	38%
10.81	51.43	40.62	2.94	0.1084	0.0240	0.2	48%
10.58	51.68	41.10	2.93	0.1083	0.0230	0.2	50%
10.34	48.87	38.53	2.87	0.1091	0.0280	0.3	41%
10.67	58.45	47.78	2.90	0.1097	0.0120	0.1	97%
10.46	55.51	45.05	2.86	0.1070	0.0180	0.2	63%
10.12	54.84	44.72	2.92	0.1096	0.0190	0.2	61%
9.89	58.74	48.85	3.60	0.7401	0.0940	0.9	83%
10.75	52.94	42.19	2.85	0.1080	0.0210	0.2	54%
9.60	59.77	50.17	5.35	1.4032	0.1560	1.5	95%

Applying the formulas presented above, the following results are obtained:

$Q_{Electric} = 0.164$ kWh; $Q_{Gas} = 7.996$ kWh; $Q_{H2O} = 5.934$ kWh; $\varepsilon = 70.6$ %.

8.1.4 – Assessment of Critical Temperatures in Heat Cell

For assessing the temperatures involved in the heat cell, it was used the adjustment seen in Table 2:

- Minimum Heat Input: 9.4 kW with CO_2 of 8.5% (v/v);
- CH Maximum Heat Input: 24.0 kW with CO_2 of 9.3% (v/v).

The temperatures under assessment were the following:

- Burner seal measured between seal and heat exchanger (see Figure 67);
- Burner hood (see Figure 68);
- Heat cell left side wall (see Figure 69);
- Heat cell front plate (measured using an IR thermal camera);
- Boiler front cover (see Figure 70).

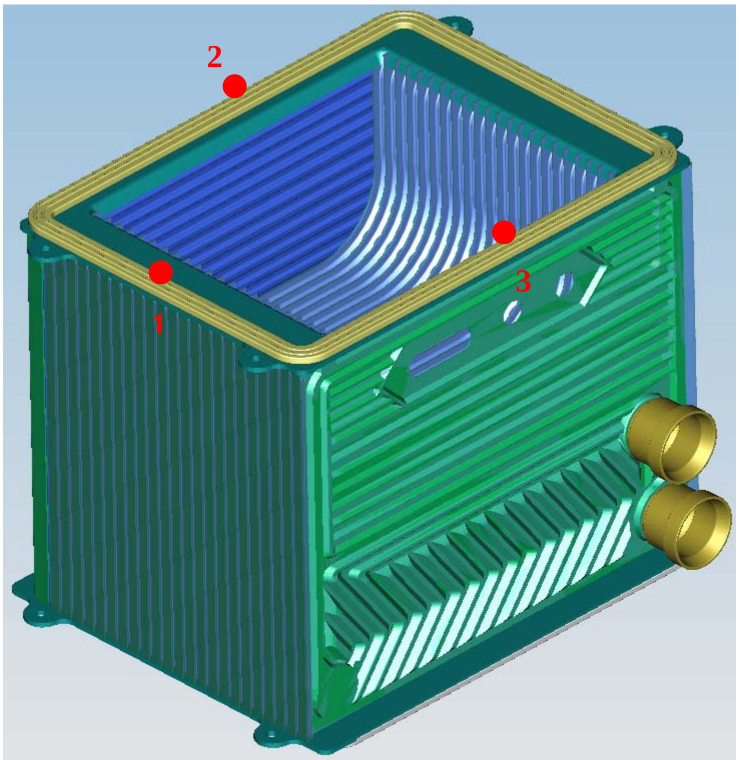

Figure 67 – Measuring points of burner seal temperature

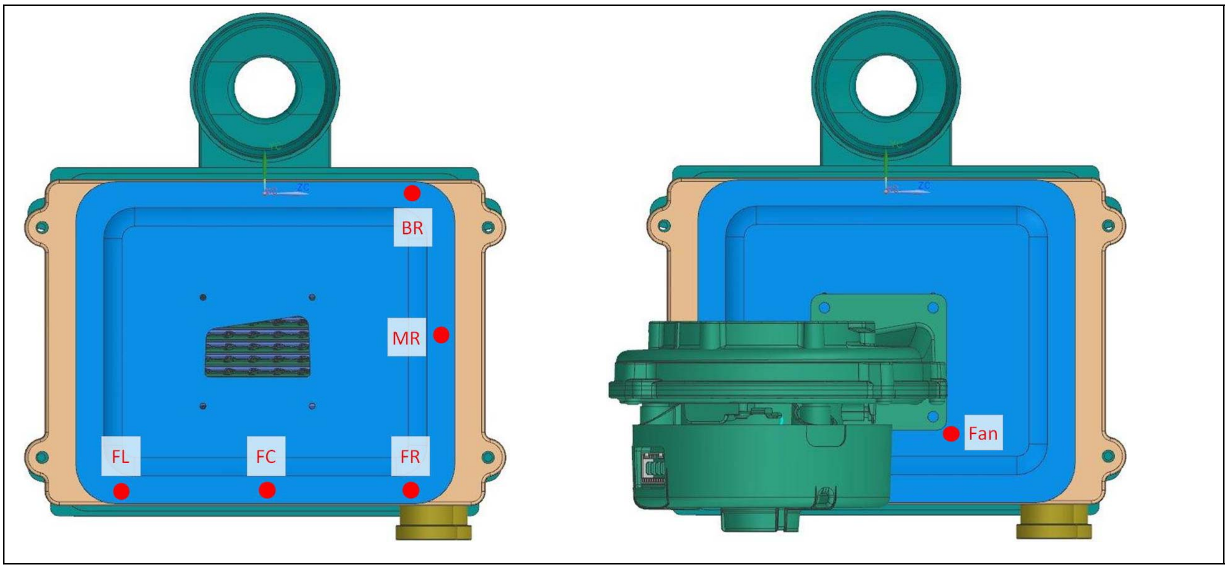

Figure 68 – Measuring points of burner hood temperatures

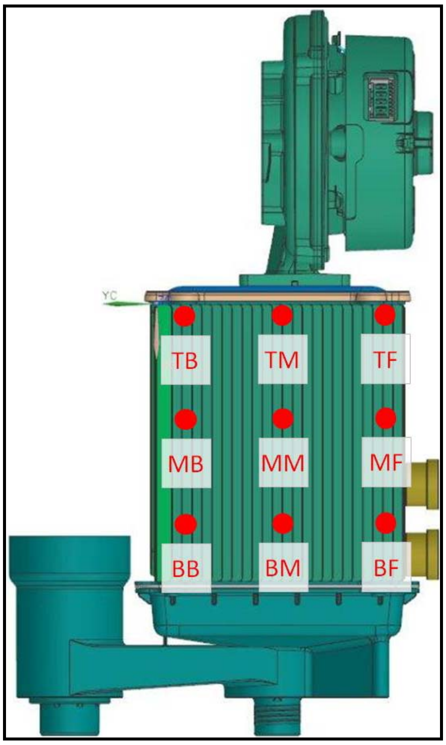

Figure 69 – Measuring points of side wall temperatures

Figure 70 – Measuring points of boiler front cover

The burner seal temperatures were evaluated since appliance start-up until steady state condition was achieved. The burner seal was registered under two different adjustments (sampling time of 1 second):

- Maximum heat input with 80/60 adjustment;
- Minimum heat input with 50/30 adjustment.

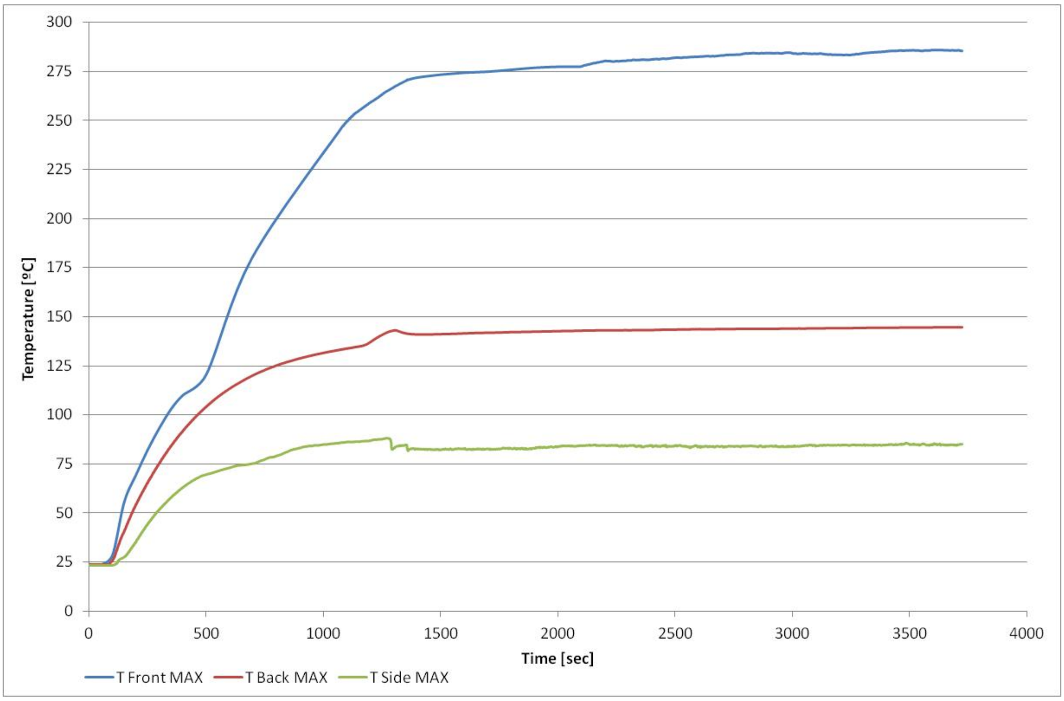

Figure 71 – Burner seal temperatures at maximum heat input

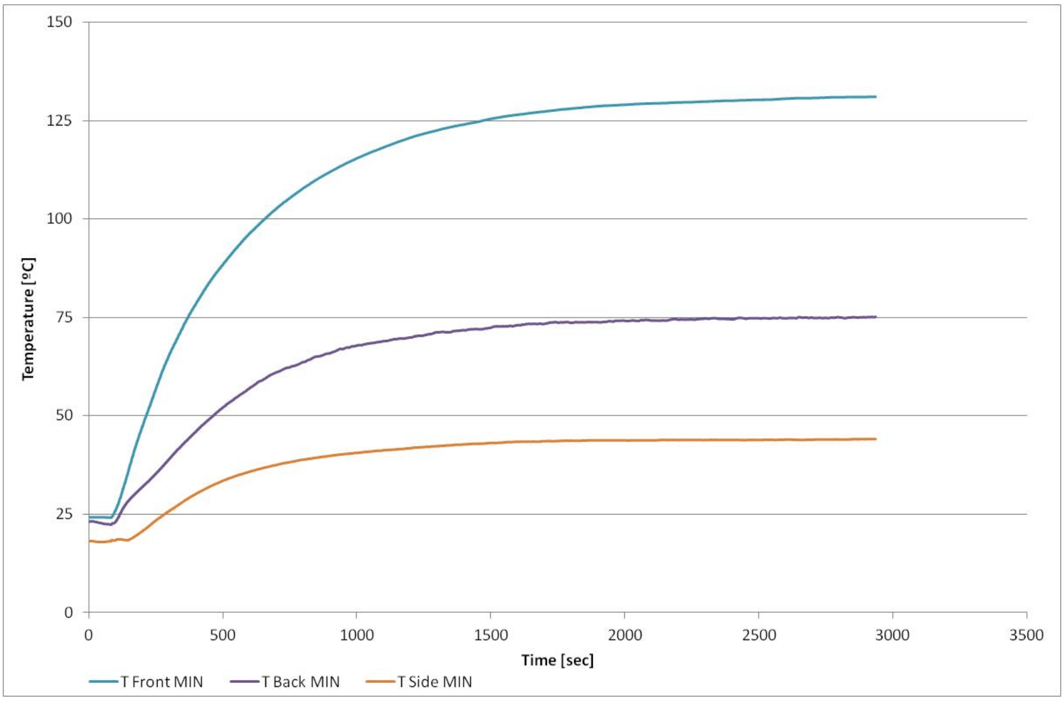

Figure 72 – Burner seal temperatures at minimum heat input

All the remaining temperatures were taken with appliance operating at maximum heat input with central heating mode adjustment of 80/60.

	CH MAX	CH MIN
FL	134 °C	126 °C
FC	123 °C	123 °C
FR	124 °C	124 °C
MR	115 °C	118 °C
BR	146 °C	145 °C
FAN	75 °C	91 °C

Table 10 – Temperature measurement results in burner hood

	CH MAX	CH MIN
TF	72.5 ºC	66.0 ºC
TM	68.0 ºC	61.0 ºC
TB	71.0 ºC	66.0 ºC
MF	55.0 ºC	-
MM	55.0 ºC	-
MB	59.0 ºC	-
BB	55.0 ºC	-
BM	57.0 ºC	-
BF	60.0 ºC	-

Table 11 – Temperature measurements results in side wall of heat exchanger

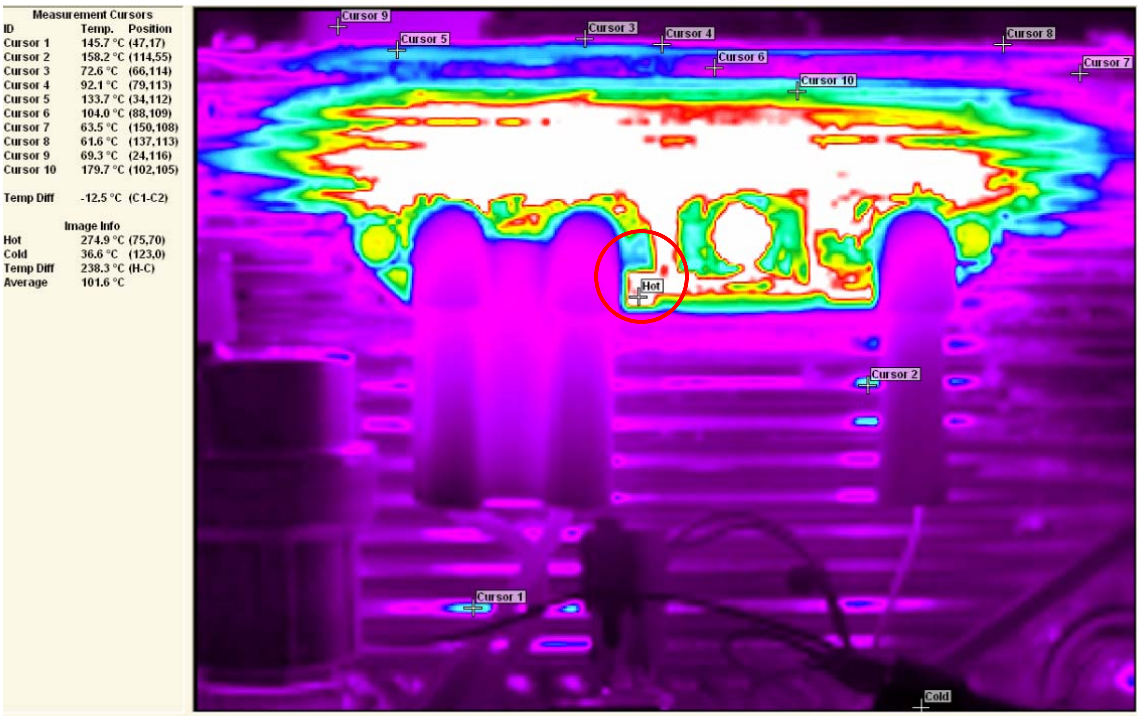

Figure 73 – Thermal measurement of front plate highlighting the hot spots: $T_{MAX} = 275$ °C

	CH MAX	CH MIN
F1	30.1 ºC	-
F2	31.5 ºC	-
F3	31.7 ºC	-
F4	30.4 ºC	-
F5	30.8 ºC	-
F6	34.2 ºC	-
F7	34.6 ºC	-
F8	32.3 ºC	-
F9	32.1ºC	-
F10	34.4 ºC	-
F11	38.4 ºC	-
F12	35.3 ºC	-
F13	31.4 ºC	-
F14	36.7 ºC	-
F15	40.0 ºC	34.0 ºC
F16	36.2 ºC	-
F17	30.0 ºC	-
F18	32.6 ºC	-
F19	34.8 ºC	-
F20	32.8 ºC	-
F21	27.5 ºC	-
F22	27.9 ºC	-
F23	28.2 ºC	-
F24	27.6 ºC	-

	CH MAX	CH MIN
R1	29.3 ºC	-
R2	29.8 ºC	-
R3	29.9 ºC	-
R4	30.6 ºC	-
R5	30.5 ºC	-
R6	30.3 ºC	-
R7	31.7 ºC	-
R8	31.4 ºC	-
R9	30.9 ºC	-
R10	32.3 ºC	29.6 ºC
R11	31.9 ºC	-
R12	31.2 ºC	-
R13	31.3 ºC	-
R14	31.1 ºC	-
R15	30.8 ºC	-
R16	29.5 ºC	-
R17	29.5 ºC	-
R18	29.6 ºC	-

Table 12 - Temperature measurements results in the appliance front cover

The air inside the appliance was 46 ºC and 41 ºC at maximum and minimum heat input respectively.

There is a clear need of reducing the temperatures in burner and ignition electrodes seals. The measured values above 275 ºC do not allow the lifetime of 15 years to be achieved. The remaining assessed temperatures are in line with expectations and pose no risk.

8.2 Sample 2

With Sample 2, the focus was given to the temperatures of heat cell, especially the front module, where Sample 1 had a poor performance. The measured points were:

- Scanning of burner seal surface with measuring points every 2 cm:
  $T_{Max\ Registered} = 80$ ºC;
- Scanning of burner hood as shown in Figure 68:
  $T_{Max\ Registered} = T_{FC} = 80$ ºC;
- Scanning of left side of heat exchanger as seen in Figure 69:
  $T_{Max\ Registered} = T_{TF} = 71$ ºC;
- Scanning of internal surface of ignition electrodes seal:
  $T_{Max\ Registered} = 82$ ºC;
- Scanning of external surface of ignition electrodes seal:
  $T_{Max\ Registered} = 180$ ºC.

8.3 Summary

The main findings of the validation presented in this chapter are summarized below:

Thermal Efficiency:

- Thermal efficiency at central heating maximum heat input of 24 kW, based only on sensible heat transfer, is 97%. This is below the project target of 98%;
- Thermal efficiency at central heating minimum heat input of 7 kW, based on sensible and latent heat transfer, is 107%. This is slightly below the project target of 107.5%, but well inside the measurement uncertainty of ± 2%;
- Domestic water tapping efficiency according cycle number 2 from EN13203-2 is 70.6% also below the project target of 72.5%.

The performance is slightly below the targets and the results achieved in simulation. The probable root cause is the uneven gas / air mixture distribution in the burner surface. By using standard gas and air components with a new burner created specifically for the laboratory validation, it is very likely to have a poor distribution in the burner surface. The simulations performed ignored this factor, as it is not the subject of the present work.

It is important to state that none of these results are of concern as they can be easily overcome. The optimization of flue gas distribution is possible to be done, starting by CFD and later using an experimental set-up with PIV measurements.

Flue Gas Emissions:

- Maximum corrected carbon monoxide emissions observed were 96 ppm (v/v) under nominal CO_2 adjustments. The project target sets a maximum of 100 ppm, which is still 10 times lower than the existing standard limits;
- Maximum corrected nitrogen oxides emissions observed were 14 ppm (v/v) (ca. 25 mg/kWh according conversion prescribed in EN 483) and therefore it is clear that the maximum NO_X Class 5 is easily achievable (NO_X weighted < 70 mg/kWh).

The targets for flue gas emissions were achieved with this preliminary design. Further improvements on carbon monoxide are possible, like:

- Improving gas / air mixture quality, optimizing the gas/air mixer;
- Changing the burning chamber volume and therefore increasing residence time.

The nitrogen oxides registered are always below 25 mg/kWh and therefore according state of the art and allow compliance with most demanding standards.

Comfort Level:

The targets of comfort are already achieved. The score of 41 points is in fact easily overcome only by decreasing the minimum flow rate of the appliance. The boiler prototype where the heat cell was installed did not allow flow detection below 4 l/min, and therefore in this specific test the score was 2 points instead of 3. By changing the water flow meter, the appliance will be able to detect water flows below 2 l/min and therefore the maximum possible score of 42 points is achievable.

Temperatures in Heat Cell:

The results with Sample 1 show very weak results in cooling of burner seal and ignition electrodes seal. Temperatures of 275 ºC were measured in both seals. The limit is set as continuous 150 ºC, the value allowed for standard elastomers used in this type of seals.

The burner hood temperatures present acceptable values, though requiring optimization, to enable temperatures below 150ºC in all possible operating conditions.

The cooling in side walls is very efficient, as the maximum temperature measured was 72.5 ºC.

The design improvements done in Sample 2 clearly solved the high temperatures measured in the burner seal, limiting them to 80 ºC under normal operation conditions. Sample 2 shows also very good improvements in the ignition electrodes seal, however limited to the surface that contacts the heat exchanger, where maximum registered temperature was 82 ºC. On the external surface contacting the ignition electrodes, the temperature reached 180 ºC, still not acceptable when considering lifetime of the seal. This high temperature is due to conduction through the stainless steel igniter pins and the ceramic. The positioning of the ignition electrodes has to be carefully analyzed. Moving this assembly to the top of the burner hood, would most likely solve this topic, as the assembly would be surrounded by gas / air mixture before burning, and therefore with temperatures limited to 80ºC.

9 Conclusions

A novel condensing heat cell concept was presented in the current dissertation. This concept is intended to provide a cost competitive solution while maintaining the most valued performance factors by the end customers. The compliance with all international harmonized standards is a must, but special national standards cannot be forgotten as the solution is not directed to any particular country.

The evaluation undertaken in this work has focused on giving a preliminary design for a low cost condensing heat exchanger. It was proven that the presented design fulfills the requirements of:

- Cost competitiveness and compactness;
- Compliance assurance of most restrictive flue gas emissions standards;
- Thermal efficiency comparable with current solutions in the market;
- Robustness and reliability (eliminating boiling and providing efficient cooling in seal connections).

The original concept could not meet the above so it was optimized using a Computational Fluid Dynamic tool. After choosing the required dimensions for providing the desired thermal efficiency, the focus was later given to:

- Eliminating boiling risk by preventing stagnant regions and water temperatures above 100 ºC;
- Enhancing heat transfer in the critical area of burner seal to allow seal temperature below 100 ºC;
- Improving cooling in the ignition electrodes area to assure the lifetime of the seal.

The concept was later prototyped, assembled in a standard condensing boiler by replacing only the heat cell, and submitted to laboratory validation where the above items were assessed.

The requirements of thermal efficiency and flue gas emissions were reached or present very low risk of not being achieved. They are in line with top performing condensing boilers existing in the market.

The comfort level achieved is excellent and with an improved water flow meter (standard component already available), the maximum score possible is achievable.

The weak point of the design is the temperature of seals, namely the ignition electrode seal. The measured value is not acceptable and the location of this assembly has to be changed to prevent this failure mode. It is proposed to change its location to the top of the burner hood, as this is an area cooled with gas / air mixture. To do so, gas / air assembly and especially the burner design have to be adapted.

The above performance factors already demonstrated the high potential of this concept for industrialization. However, additional investigation and testing work has to be carried before this important step:

1. It is mandatory to demonstrate that the robustness and reliability of the heat exchanger, especially in what regards to corrosion resistance (stress corrosion cracking, pitting, crevice and galvanic corrosion);
2. The heat cell has to be proven regarding its mechanical strength, especially to thermal stresses in abnormal operation conditions like reduced system pressure, empty system or no flow condition;
3. Specific simulation and laboratory validation for thermo-acoustic behavior is mandatory, because it is known its effect in operation noise, flue gas emissions and lifetime of materials;
4. Validation of the heat cell following the most demanding tests present in current harmonized standards is still missing.

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